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STEAM CONDENSING PLANT

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STEAM CONDENSING PLANT

IN THEORY AND PRACTICE

BY

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NEW EDITION

BLACKIE & SON LIMITED
LONDON AND GLASGOW
1926

First issued, 1925
New edition, 1926

Printed in Great Britain by
Blackie & Son, Limited, Glasgow

PREFACE

This book has been written with the object of providing students with rational methods for the design of steam condensing plant; but the author hopes that much of the subject matter will be found useful by engineers engaged in the design or operation of such plant.

In modern steam power plant, the condensing plant is so closely allied to the evaporator, the feed heater, the de-aerator, and the feed system that, to give the book some degree of completeness, a chapter on each of these latter units has been included.

An endeavour has been made to give the book some practical value by the introduction of definite experimental co-efficients and constants in the various formulæ. Some of these have been collected from the technical press, while others have been found by the author to agree with actual results.

The worked examples have an educative value and have been specially chosen to bring out the principles involved, and to exemplify methods of calculation.

The steam tables used throughout are those by Marks and Davis, published by Messrs. Longmans, Green & Co., Ltd.

Many illustrations have been included showing modern plant, and the author is indebted to the various firms mentioned in the text for permission to publish these, also to *Power* for permission to reproduce a set of curves. Every care has been taken in the text to acknowledge the source from which the information has been obtained.

The author desires to express his thanks to Messrs. G. & J. Weir, Ltd., Cathcart, Glasgow, for permission to publish data obtained in their experimental and testing departments, to Mr. J. Young for valuable hints on Water-Extraction Pumps, and to Mr. G. C. Roxburgh for the preparation of the diagrams.

The greatest care has been taken to avoid errors. Should, however, any be found, the author will be glad to have them pointed out. He will also welcome suggestions for the improvement of the book.

J. SIM.

94 LEDARD ROAD, GLASGOW.
February, 1925.

NOTE

The opportunity of a new edition has been taken to make sundry minor corrections throughout the book. The chapter on " De-aeration Plants " has been re-written so as to bring it into line with the very rapidly advancing practice on this subject.

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SYMBOLS

A	cross-sectional area of tube; entrance area to condenser.
A_{de}	area at diffuser entrance.
A_n	total area of nozzles.
a	volume passing any row of tubes.
a_a	diffuser entrance area.
a_d	diffuser outlet area.
a_t	diffuser throat area.
b	constant.
C	constant.
C_d	coefficient of discharge.
C_0	total condensation.
C_v	coefficient of velocity.
D	internal diameter of tube in inches; outside diameter of tube in inches.
d	density; internal diameter of tube in feet; pressure drop.
F	total friction loss in piping.
f	constant.
G	cooling water in gallons per minute.
G_a	condensing water for air cooling.
G_e	cooling water evaporated.
g	acceleration due to gravity.
H	total heat flow per hour; total loss in friction head.
H_E	effective head.
H_c	head lost due to condenser friction.
H_e	head lost due to entrance friction.
H_t	head lost due to tube friction.
h	heat drop; heat flow; steam head in feet.
J	Joule's mechanical equivalent of heat.
K	coefficient (eddy losses); constant; tube spacing factor.

L	latent heat.
L_w	average latent heat.
l	length of tube.
l_d	length of diffuser.
M	mass of steam condensed per second;
	total mass of air and vapour.
M_s	mass of vapour condensed out by cooler.
m	hydraulic mean depth;
	mass of air in pounds per hour;
	mass of water passing per second.
m_v	mass of vapour.
N	number of nozzles;
	number of tubes per row.
n	number of right-angled bends;
	number of tubes per flow.
P	pitch of tubes;
	Poiseuille's figure.
P_1	pressure at inlet to condenser, in pounds per square inch.
P_2	pressure at outlet of condenser, in pounds per square inch.
P_{1s}	vapour pressure at inlet.
P_{2s}	vapour pressure at outlet.
P_d	diffuser outlet pressure.
P_t	diffuser throat pressure.
p	total pressure at bottom of condenser.
p_a	partial air pressure.
p_{1a}	partial air pressure at inlet to air cooler.
p_{2a}	partial air pressure at outlet of air cooler.
p_1	absolute pressure at inlet to air cooler.
p_{2a}	absolute pressure at outlet of air cooler.
p_1	pressure at inlet to condenser, in pounds per square foot.
p_2	pressure at outlet of condenser, in pounds per square foot.
Q	quantity of steam;
	quantity of water.
Q_1	quantity of air circulating.
q	dryness fraction of steam.
R	total resistance $= R_w + R_t + R_f + R_a$.
R_1	relative humidity of entering air.
R_2	relative humidity of leaving air.
R_a	resistance due to air blanket.
R_f	resistance due to water film.
R_t	resistance across tube wall.
R_w	resistance at surface of separation of tube wall and water.
r	specific heat of air;
	total resistance to heat flow.
S	total condensing surface;
	tube surface.
S_c	condensing surface of cooler.
s	safety height of high-level condenser;
	steam temperature.

T_1	inlet water temperature.
T_2	outlet water temperature.
T_c	condensate temperature.
T_s	steam temperature.
t	temperature of air and vapour.
t_a	temperature of air leaving condenser.
t_1	temperature at inlet to air cooler.
t_2	temperature at outlet of air cooler.
t_1	temperature corresponding to V_1.
t_2	temperature corresponding to condensate temperature.
t_{1a}	inlet air temperature.
t_{2a}	outlet air temperature.
T_{Ge}	inlet temperature of G_e.
V	specific volume of steam.
V_1	vacuum at top of condenser.
V_2	vacuum at bottom of condenser.
V_a	specific volume of air.
V_s	specific volume of vapour.
V_{1a}	specific volume of air at inlet to cooler.
V_{2a}	specific volume of air at outlet of cooler.
V_{1s}	specific volume of vapour at inlet to cooler.
V_{2s}	specific volume of vapour at outlet of cooler.
V_d	diffuser discharge volume.
V_t	diffuser throat volume.
v	velocity of water in tube.
v_a	velocity of air and vapour.
v_e	critical velocity; velocity at inlet to diffuser.
v_d	velocity at diffuser discharge.
v_s	exit velocity of steam.
v_t	velocity at diffuser throat.
W	weight of steam condensed per hour; weight of water passing, in pounds per hour.
w	weight of steam condensed per square foot.
w_1, w_2	saturation weight of vapour.

γ	specific heat of air.
δ	density of steam.
θ	angle between guide cones and water jet.
θ_1	difference of temperature at cooling water inlet.
θ_2	difference of temperature at cooling water outlet.
θ_m	mean temperature difference.
μ	viscosity of water.

STEAM CONDENSING PLANT

CHAPTER I

Surface Condensers

The function of a steam condenser is to condense steam under given conditions with the least expenditure of energy and the minimum loss of heat.

There are five types of condensers in common use to-day; these are:

1. Surface condensers.
2. Low-level jet condensers.
3. High-level or barometric condensers.
4. Ejector condensers.
5. Evaporative condensers.

The most common type is the surface condenser, which has the great advantage that the condensate is not thrown to waste, but is returned to the boilers through the feed-water system. This type we shall deal with first.

The ideal to be aimed at when designing a surface condenser is to maintain a vacuum at the steam inlet corresponding to the temperature of the condensate, and to do this with a minimum pumping of air, condensate, and circulating water. The air pumping can be kept down to a minimum by the elimination of air leakage in the vacuum system; the condensate pumping depends on the conditions of the installation; and the circulating water pumping depends conjointly on the conditions of the station and quantity of water necessary, which in turn depend upon the heat flow obtainable across the tube walls.

MEAN TEMPERATURE DIFFERENCE

The heat flow across the tube walls depends upon the temperature difference between the steam on the outside of the tubes and the temperature of the water inside the tubes, and the greater the temperature difference the greater the heat flow. The true mean temperature difference is not the simple arithmetic mean.

Consider fig. 1: s is the steam temperature, which for the present may be assumed to be constant; T_1 is the inlet water temperature; T_2 is the outlet water temperature.

The water enters the tube at A and leaves at B. In travelling along the tube the temperature of the water rises as shown by the curve CD; at the inlet end, A, the temperature difference between the steam and the water is greater than the temperature difference at the outlet end, B, and hence the rate of temperature rise of the water is greater at the inlet end of the tube than it is at the outlet end. The curve CD, therefore, has its greatest slope at C, and the slope becomes progressively less from C to D.

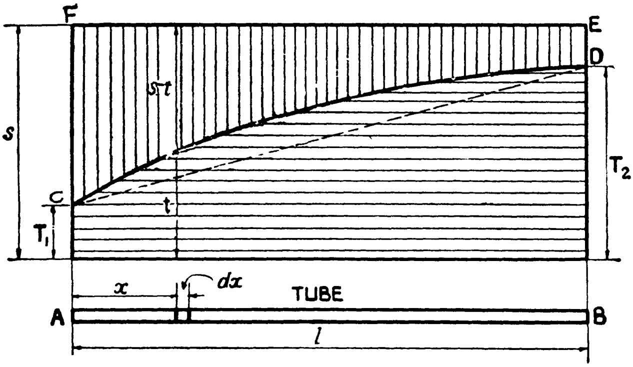


Fig. 1

Take a small element of tube length dx distant x from the inlet end of the tube, and let t be the water temperature at that point. The temperature difference at this point will be $(s - t)$.

Let W be the weight of water passing in pounds per hour. Then W will be raised in temperature in passing from x to $(x + dx)$ by an amount dt, and will absorb an amount of heat $W(dt)$ B.Th.U.

Again the heat flow across the small element of tube surface (dx) will be

$$= \frac{s - t}{R} dx.*$$

But, since the temperature is steady, the heat absorbed by the water must equal the heat flow across the tube walls.

$$\therefore W(dt) = \frac{s - t}{R} dx,$$

or

$$\frac{dt}{s - t} = \frac{dx}{WR}. \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Integrating this expression we get

$$\log_e C - \log_e(s - t) = \frac{x}{WR}, \quad \ldots\ldots\ldots\ldots\ldots (2)$$

where C is the constant of integration.

---

* See p. 7.

To find the constant of integration C we know that when $x = 0$, $t = T_1$, therefore by substitution we get

$$\log_e C - \log_e(s - T_1) = \frac{0}{WR},$$

$$\text{or} \qquad \log_e C = \log_e(s - T_1). \quad \ldots\ldots\ldots\ldots (3)$$

Substituting this in (2) we get

$$\log_e(s - T_1) - \log_e(s - t) = \frac{x}{WR},$$

$$\text{or} \qquad \log_e \frac{s - T_1}{s - t} = \frac{x}{WR}. \quad \ldots\ldots\ldots\ldots (4)$$

This equation gives the temperature of water at any point along the tube, and from it the curve CD can be plotted for any given condition.

From the fundamental equation of logarithms, viz.:

$$a^x = b,$$

where x is the log of b to the base a, the above equation (4) may be written in the following form:

$$e^{\frac{x}{WR}} = \frac{s - T_1}{s - t},$$

$$\text{or} \qquad s - t = (s - T_1)e^{-\frac{x}{WR}}. \quad \ldots\ldots\ldots\ldots (5)$$

The mean temperature difference from A to B is the difference between s and the mean of t; if, therefore, we denote the mean temperature difference by θ_m, and the length of the tube by l, we have

$$\text{area of FEDC} = \theta_m \times l.$$

$$\text{Now} \qquad \text{FEDC} = \int_0^l (s - t)dx.$$

$$\therefore \theta_m = \frac{1}{l}\int_0^l (s - t)dx.$$

Substituting from (5) the value of $(s - t)$ we get

$$\begin{aligned}\theta_m &= \frac{1}{l}\int_0^l (s - T_1)e^{-\frac{x}{WR}}dx \\ &= \frac{s - T_1}{l}\int_0^l e^{-\frac{x}{WR}}dx \\ &= \left[\frac{s - T_1}{l}e^{-\frac{x}{WR}}(-WR)\right]_0^l \\ &= \frac{s - T_1}{l}(-WR)(e^{-\frac{l}{WR}} - 1).\end{aligned}$$

From (5) we have

$$s - t = (s - T_1)e^{-\frac{x}{WR}}.$$

Also when $x = l$, $t = T_2$,

$$\text{or} \qquad s - T_2 = (s - T_1)e^{-\frac{l}{WR}}.$$

Then, substituting these values in the above equation, we get

$$\theta_m = -\frac{WR}{l}[(s - T_2) - (s - T_1)]$$

$$= \frac{WR}{l}(T_2 - T_1) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (6)$$

It may be noted that this equation can be deduced at once from equation (1) by integration. For $WRdt = dx(s - t)$, so that

$$WR\int_{T_1}^{T_2} dt = \int_0^l (s - t)dx, \text{ i.e. } WR(T_2 - T_1) = l\theta_m.$$

Again from equation (4) we have, when $x = l$, $t = T_2$,

$$\log_e\left(\frac{s - T_1}{s - T_2}\right) = \frac{l}{WR}.$$

Then, substituting this value in equation (6), we get

$$\theta_m = \frac{T_2 - T_1}{\log_e\left(\frac{s - T_1}{s - T_2}\right)}$$

This equation is more convenient for practical use when written in the form:

$$\theta_m = \frac{(s - T_1) - (s - T_2)}{\log_e\left(\frac{s - T_1}{s - T_2}\right)}.$$

Now $(s - T_1)$ is the difference of temperature at cooling-water inlet; call this θ_1. $(s - T_2)$ is the difference of temperature at cooling-water outlet; call this θ_2.

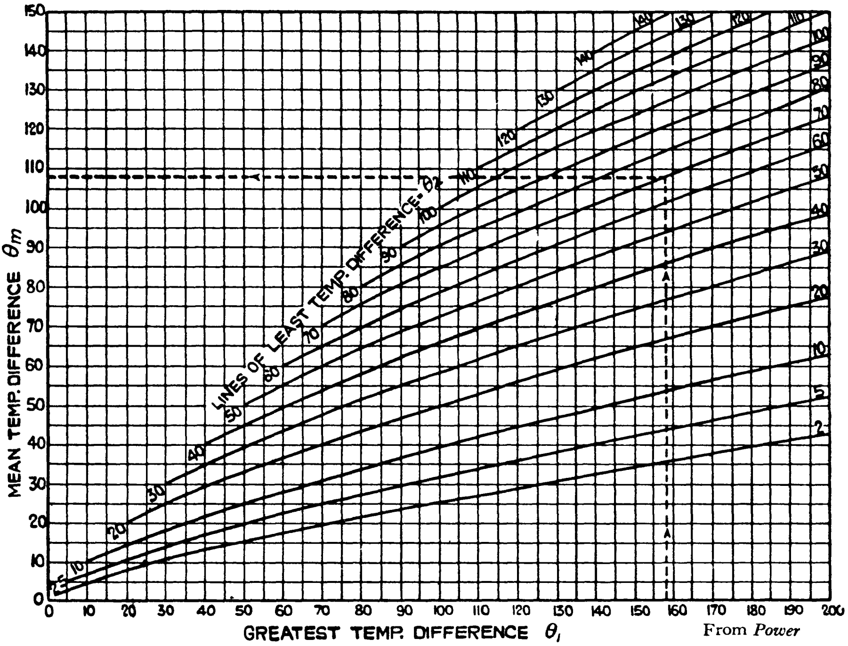
The equation then takes the form:

$$\theta_m = \frac{\theta_1 - \theta_2}{\log_e\frac{\theta_1}{\theta_2}} \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

$$\text{or} \qquad \theta_m = \frac{\theta_1 - \theta_2}{2{\cdot}3 \log_{10}\frac{\theta_1}{\theta_2}} \ldots\ldots\ldots\ldots\ldots (8)$$

The above is Grashof's equation, and has been here deduced for a constant steam temperature, but the same equation holds when the steam temperature at outlet is lower than that at inlet. In general $s - T_2$ is the difference between incoming steam and outgoing cooling water, and $s - T_1$ is the difference of temperature between the condensate and incoming cooling water. The equation also holds whether the flow be counter current or cross current or parallel current, and is also independent of the number of passes. Further, it is applicable to feed water-heaters, oil coolers, or the like.

To facilitate the calculation of this equation many charts have been prepared, the most straightforward one which has come under the author's observation being that published in *Power*, Vol. 55, No. 11, which is reproduced in fig. 2.



From *Power*

Fig. 2

To use the chart: *first*, find the greatest temperature difference, θ_1, of the fluids, which in a steam condenser is usually the temperature at the cooling-water inlet; *second*, find the least temperature difference θ_2; *third*, enter the chart at the bottom at the greatest temperature difference θ_1 and move vertically till the curve of least temperature difference, θ_2, is reached; *fourth*, move horizontally to the left-hand scale and read there the mean temperature difference θ_m.

Example.—A surface condenser of the two-flow type has cooling water entering at 60° F. and leaving at 75° F.; the temperature of the steam entering the condenser is 83° F., and the temperature of the condensate leaving the condenser is 78° F. Find the mean temperature difference.

From equation (8) $\theta_m = \dfrac{\theta_1 - \theta_2}{2\cdot3 \log_{10}\dfrac{\theta_1}{\theta_2}}$.

$$\theta_m = \frac{(78 - 60) - (83 - 75)}{2\cdot3 \times \log_{10}\dfrac{78 - 60}{83 - 75}}$$

$$= \frac{18 - 8}{2\cdot3 \times \log_{10}2\cdot25}$$

$$= \frac{10}{2\cdot3 \times 0\cdot352}$$

$$= 12\cdot35^\circ \text{ F.}$$

If we compare this with the arithmetic mean we find the latter to be

$$\theta_{\text{AM}} = \frac{(78 - 60) + (83 - 75)}{2}$$

$$= \frac{18 + 8}{2}$$

$$= \frac{26}{2}$$

$$= 13^\circ \text{ F.}$$

The dotted line in fig. 1 shows how the temperature along the tube would rise should the arithmetic mean be the correct one, but, as stated on p. 2, the temperature difference at the inlet is greatest, and hence the water temperature at the inlet end has a greater rate of rise than at the outlet, and the true mean temperature difference is therefore less than the arithmetic mean.

Grashof's equation is not strictly true when the steam which is being condensed contains air, and the greater the percentage of air the greater the deviation from the above equation. There is also the change in the viscosity of the cooling water which affects the truth of the equation.

In a steam condenser the air density varies, being greatest at the coldest part, that is, where the cooling water enters. As will be explained later, this air tends to form a blanket round the tubes and retard the heat flow, and the denser this blanket, the greater the resistance to heat flow. The heat, therefore, added to the circulating water at the bottom of a modern condenser is less than the above equation takes account of, so that the rate of rise of the curve CD is less at the inlet end C and greater at the outlet end D than is shown in fig. 1.

The above equation (8) for θ_m is generally accepted, and all the constants which follow have been worked out assuming the equation to hold. This does not affect the final results since the constants are deduced from experiment; moreover, this equation gives a lower θ_m than the actual, which is on the safe side when fixing up the actual surface necessary for any condenser.

HEAT FLOW ACROSS TUBE WALLS

There have been many formulæ deduced for the heat flow across the tube walls of steam condensers, but the theory first published in *Engineering*, January, 1914, has been found by the author to agree best with tests on actual steam condensers. In what follows, therefore, some subject matter has been drawn from the above journal.

The laws governing the flow of heat across a metal wall or partition are:

1. The heat flow varies directly as the temperature difference between the one side and the other.
2. The heat flow varies inversely as the resistance to the heat flow.

Let *ab* (fig. 3) be a metal wall, with a fluid of temperature s on the one side and a fluid of temperature t on the other side. If r be the total resistance to heat flow from the one fluid to the other, and h the heat flow, then

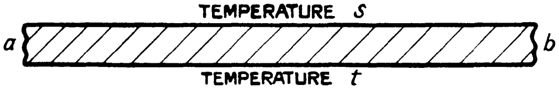


Fig. 3

$$h \propto \frac{s-t}{r},$$

or $$h = C\left(\frac{s-t}{r}\right),$$

where C is a constant and can be incorporated in r, so that we can write for $\frac{C}{r}$ simply $\frac{1}{R}$. Hence

$$h = \frac{s-t}{R} \qquad (9)$$

In a steam condenser the resistance R is made up of four resistances, viz:

R_w, the resistance at the surface of separation of the tube wall and the water:

R_t, the resistance across the tube wall itself:

R_f, the resistance due to the water film of condensed steam on the outside of the tube:

R_a, the resistance due to the air blanket surrounding the tube.

Fig. 4 indicates these resistances.

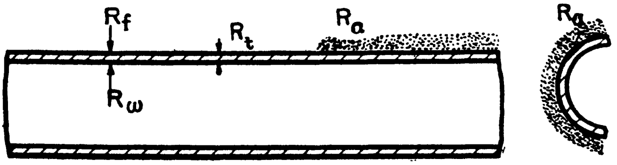


Fig. 4

These resistances depend on the following factors:

R_w depends upon the water velocity in the tube, upon the tube diameter, and upon the viscosity and density of the water passing through the tube.

R_t depends upon the thickness and upon the material from which the tube is made, and is constant for any one tube.

R_f depends upon the viscosity and density of the water film adhering to the outside of the tube.

R_a depends upon the quantity of air in the condenser, an air blanket forming round the tube due to the steam condensing and leaving the air behind, thus preventing fresh steam from getting at the tube; this resistance is by far the greatest one.

It is noteworthy that these resistances exist mainly at surfaces of separation, the resistance across the tube wall being comparatively small

If R be the total resistance from the steam to the water, then

$$R = R_a + R_f + R_t + R_w. \qquad \ldots\ldots\ldots\ldots\ldots (10)$$

When equation (9), viz. $h = \frac{s - t}{R}$, is applied to the conditions which obtain in a steam condenser, the numerator $(s - t)$ is the mean temperature difference between the steam on the one side of the tube wall and the water on the other side.

Denoting the total heat flow per hour across the condensing surface by H, we get

$$H = \frac{\theta_m}{R},$$

or

$$R = \frac{\theta_m}{H}.$$

Now the total heat flow $H = W \times L$, where W is the weight of steam condensed per hour, and L the heat given up by 1 lb. of steam in condensing.

$$\therefore R = \frac{\theta_m}{W \times L}.$$

Now, instead of taking the whole weight of steam condensed, let us take the weight w condensed per square foot of tube surface, so that

$$R = \frac{\theta_m}{w \times L}. \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

This gives a value to R which is very small, and to facilitate practical calculation it is better to change the unit of resistance so that in the new units

$$R = \frac{1000 \times \theta_m}{w \times L}, \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (12)$$

or

$$w = \frac{1000 \times \theta_m}{R \times L}.$$

For practical work L may be taken at 1000 B.Th.U., so that

$$w = \frac{\theta_m}{R_w + R_t + R_f + R_a} \cdots\cdots\cdots\cdots\cdots (13)$$

We have here now an equation which enables us to determine the rate of condensation (w), if we know the practical value of the individual resistances R_w, &c. Let us first consider the resistance R_w.

It is now fairly well established that this resistance R_w varies with the viscosity, density, and velocity of the cooling water and the internal diameter of the tube. From experiments and investigations made by the author on actual condensers this is borne out in practice.

This resistance

$$R_w = \frac{\frac{\mu}{d}k}{v^{0\cdot82}},$$

where μ = viscosity of the water, d = density, v = velocity of water in tube in feet per second, and k = a constant depending on the diameter of the tube.

For any one diameter of tube we can write:

$$\frac{\mu k}{d} = C,$$

and $$R_w = \frac{C}{v^{0\cdot82}} \cdots\cdots\cdots\cdots\cdots\cdots\cdots (14)$$

The value of C has been found from experiment, and may be read directly from the graphs in fig. 5 for the diameter of tube used. The water temperature to be taken is the average between the inlet and the outlet, and v can be found from the quantity of water passing.

The tube velocity should always be kept above the critical. Below the critical velocity the water flows through the tube in stream lines, and the transfer of heat from the tube walls to the water is simply by conduction, and water being a very bad conductor the heat transference becomes very small. Above the critical velocity turbulent flow exists, the water swirling and eddying throughout the whole tube. Each particle of water, therefore, remains in contact with the tube wall for a small fraction of a second, picks up its quota of heat, and is then swirled away through the whole body of water in the tube. The heat, therefore, is quickly transferred to the water in the tube, and the greater the tube velocity the greater the turbulence, and hence the faster the heat is imparted to the water.

As equation (14) shows, the resistance to the heat flow R_w varies inversely as the velocity raised to the power 0·82.

The value of R_t, for clean condenser tubes 18 S.W.G. thick, may be taken as 0·08; a very small deposit, however, on the tube surface is sufficient to considerably increase this value, as scale is a very bad heat conductor.

With the high vacuum demanded to-day by turbine conditions, it is very essential that the tubes be kept free from scale or oily deposits, because of the small available temperature gradient across the tube wall, and the smallest additional resistance to the heat flow is immediately shown in the vacuum. This is one of the advantages of a high water velocity condenser, as the high velocity not only decreases the resistance R_w, but it tends to scour the tubes of these deposits. The author has had occasion to examine condensers which were reported to be giving a vacuum below the designed value, and frequently the removal of these thin deposits by the application of a weak solution of hydrochloric acid was all that was necessary to bring the vacuum over the designed value.

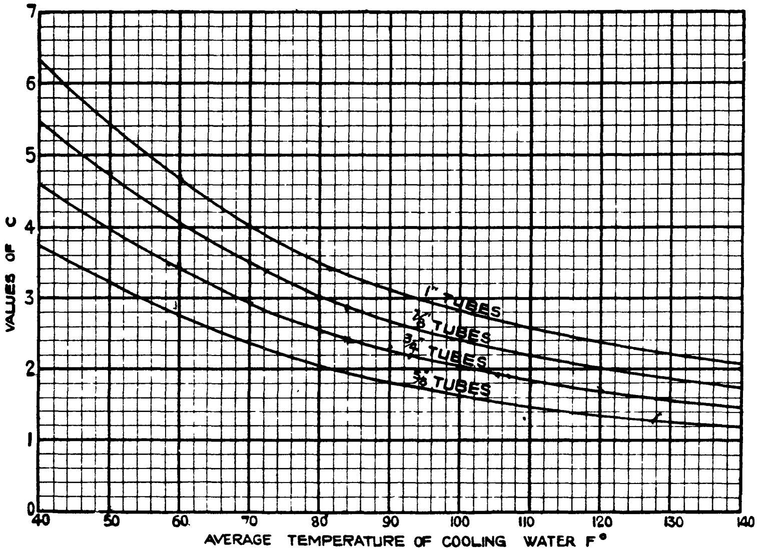


Fig. 5

The value of R_f is not, so far as the author is aware, accurately known, but the sum of R_f and R_a is known with sufficient accuracy for practical purposes, and varies with the design of the condenser and the conditions under which it operates.

Minimum and maximum values of $(R_f + R_a)$ are:

turbines, 0·6 to 1·2;
reciprocating engines, 1·2 to 1·5.

The value of R_a for air-free steam has been investigated by Professor H. L. Callendar, F.R.S., and others. Callendar's value for R_a (see *Encyclopædia Britannica*, article " Heat ") works out for the units chosen here as 0·342. This was for steam at atmospheric pressure and 212° F. temperature; but, for steam at condenser pressures, this should be increased to 0·48. However, since it is impossible under condenser conditions to get absolutely air-free steam, the true value of R_a need not concern us here. Moreover the percentage of air in the steam varies very considerably from the top to the bottom of a surface condenser, due to the steam condensing and leaving the air behind, so that the air density is greatest where the condenser is coldest, or, in the present-day surface condensers, at the bottom. Therefore, for our present practical needs we only require the average value of R_a.

We are now in a position to fix up the necessary condensing surface under any specified conditions, and this we proceed to do.

Example.—A land surface condenser of the two-flow type has cooling water entering at 50° F. and leaving at 70° F., the temperature of the steam entering the condenser is 80° F., and the temperature of the condensate leaving the condenser 73° F.; the tube velocity is 6 ft. per second, and the tubes are $\frac{3}{4}$ in. O/D., 18 S.W.G. thick. It is required to find the possible rate of condensation and the surface necessary to condense 150,000 lb. of steam per hour. Take $L = 1000$ B.Th.U. per pound, and $R_a = 1{\cdot}15$.

First find the mean temperature difference by equation

$$\theta_m = \frac{\theta_1 - \theta_2}{2{\cdot}3 \log_{10}\frac{\theta_1}{\theta_2}}$$

$$= \frac{(73-50)-(80-70)}{2{\cdot}3 \log_{10}\frac{73-50}{80-70}}$$

$$= \frac{23-10}{2{\cdot}3 \times \log_{10}2{\cdot}3}$$

$$= \frac{13}{2{\cdot}3 \times 0{\cdot}362},$$

$$\theta_m = 15{\cdot}62^\circ \text{ F.}$$

Then by equation (13) and making $R_f + R_a = R_a$ and substituting we get

$$w = \frac{\theta_m}{R_w + R_t + R_a}$$

$$= \frac{15{\cdot}62}{R_w + 0{\cdot}08 + 1{\cdot}15}.$$

Now from the curves in fig. 5 read off the value of C for $\frac{3}{4}$-in. O/D tubes, and for the average temperature of the cooling water which will be 60° F.

Then $$R_w = \frac{C}{v^{0{\cdot}82}}$$

$$= \frac{3{\cdot}4}{6^{0{\cdot}82}}$$

$$= \frac{3{\cdot}4}{4{\cdot}348} = 0{\cdot}782.$$

Substituting this value above we get

$$w = \frac{15{\cdot}62}{0{\cdot}782 + 0{\cdot}08 + 1{\cdot}15}$$
$$= \frac{15{\cdot}62}{2{\cdot}012}$$
$$= 7{\cdot}76 \text{ lb. per square foot.}$$

Therefore to condense 150,000 lb. per hour the surface

$$S = \frac{150{,}000}{7{\cdot}76} = 19{,}330 \text{ sq. ft.}$$

Example.—A marine surface condenser, having 8000 sq. ft. of surface and $\frac{3}{4}$-in. O/D brass tubes, 18 S.W.G. thick, is to condense 75,000 lb. of steam per hour. Find the vacuum which will be attained at the top of the condenser, when the inlet and outlet temperatures of the cooling water are 60° F. and 80° F. respectively. The temperature of the condensate is 10° F. lower than the vacuum temperature at the top of the condenser. Assume that $L = 1000$ B.Th.U., and $R_a = 0{\cdot}9$. The tube velocity $v = 6$ ft. per second.

Answer, 28·5 in. Hg.

APPLICATION OF DALTON'S LAW OF PARTIAL PRESSURES IN STEAM CONDENSERS

Dalton's law of partial pressures states that in any mixture of gases the total pressure exerted is the sum of the pressures of each of the gases which make up the mixture.

In a steam condenser we have a mixture of steam and air, and the total absolute pressure which exists in the condenser is the sum of the pressures exerted by the steam and the non-condensable gases. These gases consist chiefly of air, CO_2, and free oxygen: the CO_2 and O are extremely small quantities in comparison with the air, and may be neglected. We therefore have: $p = p_a + p_s$; where p is the total pressure; p_a, the air pressure; and p_s, the steam pressure.

Now p_a and p_s vary throughout the condenser from the steam inlet to the air-suction branch. At the inlet the air pressure is a small proportion of the total pressure, while at the air-pump suction the air pressure is a large proportion of the total pressure. These relative pressures of air and steam depend on the percentage of air present in the steam entering the condenser. At the inlet the air pressure is so very small that it may be neglected, so that, if the vacuum or absolute pressure at the inlet is measured, the temperature may be taken to be that corresponding to the vacuum.

We here assume that the entering steam is not superheated, as with the modern large turbine the expansion is carried too far for the superheat to reach the exhaust. In small turbines, which are sometimes used for driving auxiliaries, only single wheels are used. When these are supplied with superheated steam the exhaust is generally superheated, so that for these cases the above assumption is not true. We shall, however, for the present confine ourselves to the large machine. Assuming then that the entering steam is not superheated, the temperature and pressure are corresponding and are as given in the steam tables. As the steam flows through the condenser it progressively condenses and cools and falls in pressure. The air which is carried along with the steam, at the same time cools and rises in pressure, so that, as the steam pressure falls, the air pressure rises. At any point in the flow, however, the sum of the air and steam pressures is always equal to the total pressure which obtains at that point. Further, there is a drop in total pressure from the top to the bottom of the condenser due to the initiation of the entering steam velocity. This point, however, we shall investigate later. At the bottom of the condenser the steam will all have condensed, leaving its vapour at a pressure which corresponds to the temperature of the water, and the air pressure will be the difference between the total pressure and the vapour pressure. These conditions are shown in fig. 6.

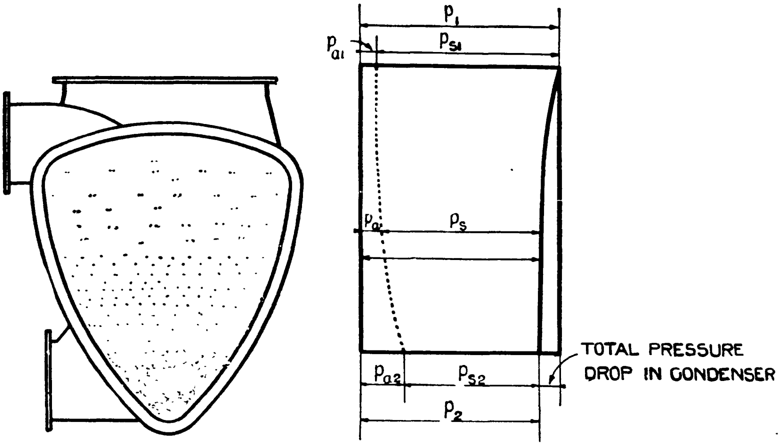


Fig. 6.—Pressure Conditions in Condenser

The dots indicate how the air density increases as the steam condenses. It should be clearly borne in mind that the air-suction branch in a modern surface condenser is taken from the bottom, because the air density is greatest there, and the air suction should always be taken from the coldest point. An example of the air suction being taken from the top of the condenser is to be found in some jet condensers, where the steam enters at the bottom and the water enters at the top. The water rains down over suitably arranged trays, while the steam passes upwards through this rain of falling water. The air, naturally, is coldest at the top, and the air suction is therefore taken from this point. The fact of the air density being greatest at the coldest point is accentuated by the steam velocity in the condenser, the steam carrying the air with it.

Example.—In a steam condenser the measured vacuum at the top of the condenser was 29 in. of mercury corrected to a 30-in. barometer, and at the bottom of the condenser 29·2 in. Hg. The temperature of the condensate was 65° F. What was the partial air pressure at the bottom of the condenser?

From the steam tables we find that the vapour pressure which corresponds to 65° F. is 0·304 lb. per square inch absolute, which is equivalent to a vacuum of 29·38 in. Hg.

$$\begin{aligned} \therefore p_s &= 30 - 29{\cdot}38 \\ &= 0{\cdot}62 \text{ in. Hg.} \end{aligned}$$

The absolute pressure at the bottom of the condenser

$$\begin{aligned} p &= 30 - 29{\cdot}2 \\ &= 0{\cdot}8 \text{ in. Hg.} \end{aligned}$$

$$\begin{aligned} \text{Now} \qquad p &= p_a + p_s. \\ \therefore p_a &= 0{\cdot}8 - 0{\cdot}62 \\ &= 0{\cdot}18 \text{ in. Hg.} \end{aligned}$$

This can be more simply found by taking the difference between the vacuum corresponding to the condensate temperature and the vacuum which obtains at bottom of the condenser,

$$\begin{aligned} \text{i.e.} \qquad p_a &= 29{\cdot}38 - 29{\cdot}2 \\ &= 0{\cdot}18 \text{ in. Hg.} \end{aligned}$$

The pressure drop or vacuum drop across the condenser

$$\begin{aligned} &= 29{\cdot}2 - 29 \\ &= 0{\cdot}2 \text{ in. Hg.} \end{aligned}$$

QUANTITY OF COOLING WATER AND POSSIBLE VACUUM

The vacuum which it is possible to attain depends upon two factors.

1. The inlet temperature of the cooling water.
2. The quantity of cooling water passing through the condenser.

It is usual to design a condenser for what is known as the economical vacuum, and this depends upon the condition of the whole installation, e.g. the cost of pumping the cooling water, the reduced operating costs consequent on the high vacuum and the initial cost of the plant. If the supply of cooling water be plentiful and the cooling water system arranged with a siphonic action, then it usually pays to design the condenser with a temperature difference between the outgoing cooling water and the entering steam of about 8° F. This gives a large condenser with high cooling water charges, but the reduced steam consumption of the main turbine, consequent on the high vacuum, usually warrants the increased cost. On board ship, where space is valuable, a smaller condenser is usually installed than would be for a land installation. This is not done solely on account of the space taken up by the condenser itself, but by the space taken up by the large low-pressure turbine.

The following are typical normal vacua:

Light craft, e.g. torpedo-boats, 27 in. Hg, sea temperature, 50° F.
Battle cruisers and battleships, $28\frac{1}{2}$ in. Hg.
Merchant ships, $28\frac{1}{2}$ in. Hg.
Modern land power stations, 29 to $29\frac{1}{4}$ in. Hg (cooling water taken from river); 28 to $28\frac{1}{2}$ in. Hg (cooling water taken from cooling tower).

Let V_1 be the vacuum at the top of the condenser;
V_2, the vacuum at the bottom of the condenser;
t_1, the temperature corresponding to V_1;
t_2, the temperature of the condensate leaving the condenser;
T_1, the inlet temperature of the cooling water;
T_2, the outlet temperature of the cooling water.

Then the difference between the ingoing steam temperature and the outgoing cooling water

$$= t_1 - T_2.$$

Good average values of this difference for various installations are:

Land power stations	8° to 12° F.
Torpedo-boats	40° to 50° F.
Battleships and cruisers	15° to 20° F.
Merchant vessels (turbines)	12° to 15° F.
Merchant vessels (reciprocating)	25° to 30° F.

The difference between the ingoing steam temperature and the condensate temperature $(t_1 - t_2)$ is about 5° to 8° F. for condensers for turbines, and about 10° to 15° F. for condensers for reciprocating engines. We shall later investigate why there is this difference between turbine and reciprocating-engine condensers.

To find the quantity of cooling water necessary to condense a given quantity of steam is a problem of a simple heat exchange.

The exhaust steam entering a condenser is usually slightly wet, even when the steam supplied to the main turbine is highly superheated. The steam in condensing gives up its latent heat and the sensible heat due to the cooling of the condensate below the temperature corresponding to the vacuum; this latter effect is due to the air which leaks into the condenser. The cooling water absorbs the heat given up by the steam, less the heat lost by radiation. This loss is a very small quantity and may be neglected. The specific heat of the water is usually taken as 1.

Let G be the quantity of cooling water in gallons per minute;
W, the weight of steam condensed per hour in pounds;
q, the dryness fraction of the steam.

Then the heat given up per hour by the steam in condensing

$$= W(qL + t_1 - t_2).$$

The heat absorbed per hour by the cooling water

$$= 600\ G(T_2 - T_1). \qquad \ldots\ldots\ldots\ldots (15)$$

These two quantities are equal.

$$\therefore\ 600\ G(T_2 - T_1) = W(qL + t_1 - t_2),$$

$$\text{or} \qquad G = \frac{W(qL + t_1 - t_2)}{600\ (T_2 - T_1)} \text{ gall. per minute.} \qquad \ldots\ldots\ldots\ldots (16)$$

For present-day conditions under which steam turbines are operated

$$(qL + t_1 - t_2) = 1000 \text{ approximately,}$$

so that very approximately

$$G = \frac{10W}{6(T_2 - T_1)}. \qquad \ldots\ldots\ldots\ldots\ldots\ldots (17)$$

Example.—In testing a steam condensing plant the following data were taken. Find the dryness fraction of the steam entering the condenser.

Vacuum at top of condenser	28 in. Hg.
Temperature of condensate	92° F.
Inlet temperature of cooling water	75° F.
Outlet temperature of cooling water	90° F.
Quantity of cooling water	15,400 gall. per minute.
Quantity of condensate	140,000 lb. per hour.

The temperature of the steam entering the condenser will be that corresponding to a vacuum of 28 in., which, from the steam tables, is 101° F., so that

$G = 15{,}400$ gall. per minute.
$W = 140{,}000$ lb. per hour.
$T_1 = 75^\circ$ F.
$T_2 = 90^\circ$ F.
$t_1 = 101^\circ$ F.
$t_2 = 92^\circ$ F.
$L = 1035{\cdot}1$ from the steam tables.
q is unknown.

Inserting these values in equation (16) we get

$$15{,}400 = \frac{140{,}000\ (q \times 1034{\cdot}6 + 101 - 92)}{600\ (90 - 75)},$$

$$\text{i.e.} \qquad q = \frac{\dfrac{15{,}400 \times 600 \times 15}{140{,}000} - 9}{\dfrac{1035{\cdot}1}{1}}$$

$$= \frac{990 - 9}{1035{\cdot}1}$$

$$= 0{\cdot}948.$$

Example.—In considering the design of a certain power station it was required to condense 100,000 lb. of exhaust steam per hour. The available quantity of circulating water was 8000 gall. per minute, at a temperature of 60° F. It was found from consideration of cost that a temperature difference, between the entering steam and the outgoing circulating water, of 8° F. could be economically allowed. Calculate what the vacuum would be, assuming that there were 1000 B.Th.U. of heat to be extracted per pound from the steam.

From equation (17) the outlet temperature of the cooling water will be:

$$8000 = \frac{10 \times 100{,}000}{6(T_2 - 60)}.$$

$$\therefore T_2 = \frac{1{,}000{,}000}{6 \times 8000} + 60$$

$$= 20{\cdot}82 + 60$$

$$= 80{\cdot}82^\circ \text{ F.}$$

The temperature of the entering steam will therefore be $80{\cdot}82 + 8 = 88{\cdot}82^\circ$ F., and from the steam tables the vacuum corresponding to this temperature is 28·64 in. of mercury.

LOSS OF HEAD DUE TO FRICTION IN CONDENSER TUBES

The loss of head due to the fluid friction of the cooling water passing through the tubes, &c., is a running charge, and should therefore be reduced to the minimum consistent with the necessary heat flow across the tube walls.

This friction loss consists of three parts.

1. The loss due to the tube itself, and known as the *tube friction.*
2. The loss at the entry end of the tube, and known as the *entrance friction.*
3. The loss in the water boxes due to the reversal of the flows, &c., and known as the *condenser friction.*

If H be the total head lost, in feet, composed of the parts
H_t, due to tube friction,
H_e, due to entrance friction,
H_c, due to condenser friction,
then $H = H_t + H_e + H_c$.

If G is the quantity of water passing through each tube in gallons per minute, l the length of each tube in feet, and D the internal diameter of the tubes in inches, then

$$H_t = \frac{G^2 l}{3(3D)^5} \quad \text{.....................(18)}$$

$$H_e = \left(\frac{G}{13D^2}\right)^2 \quad \text{.....................(19)}$$

If the condenser be of the two-flow type then the total tube friction will be $2H_t$, and the total entrance friction $2H_e$; similarly, if the condenser be of the three-flow type these friction losses will be $3H_t$ and $3H_e$ respectively, and so on for any number of flows.

The above formulæ are those due to Box, and have been found by the author to give results in close agreement with what obtains in actual condensers.

If v be the tube velocity in feet per second, then equation (18) becomes

$$H_t = 0{\cdot}005724\left(\frac{v^2 l}{D}\right), \dots\dots\dots\dots\dots\dots (20)$$

and equation (19) becomes

$$H_e = 0{\cdot}02469\, v^2. \dots\dots\dots\dots\dots\dots\dots (21)$$

In fig. 7 the author has plotted the tube and entrance friction for tube velocities varying from 3 to 11 ft. per second, and for tubes of $\frac{5}{8}$ in., $\frac{3}{4}$ in., $\frac{7}{8}$ in., and 1 in. outside diameter, having a thickness of 18 S.W.G., and a length of 10 ft.

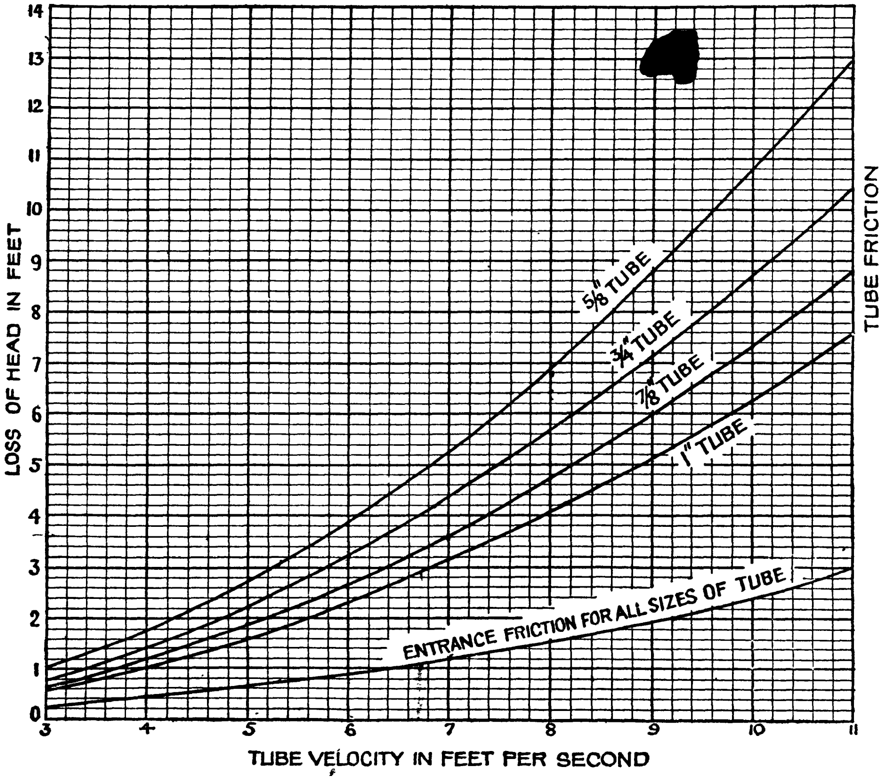


Fig. 7.—Friction Head in 10 ft. Length Condenser Tubes

These curves are to facilitate calculation, and are very handy when determining the friction loss in any condenser.

It is interesting to note that the entrance friction depends entirely on the velocity and is independent of the tube diameter.

The value of the condenser friction H_c depends upon the design of the water boxes and the number of flows or passes.

Good average values for H_c are:

	Feet.	Inches.
For a single-flow condenser	1	0
For a two-flow condenser	1	6
For a three-flow condenser	2	0
For a four-flow condenser	2	6

Example.—A surface condenser of the three-flow type has 3500 $\frac{3}{4}$-in. O/D tubes, 18 S.W.G. thick, 9 ft. 6 in. long. The quantity of cooling water passing is 6800 gall. per minute. Calculate the total loss in friction head from inlet to outlet.

First find the tube velocity.

$$v = \frac{\text{[illegible]tity of water passing in cubic feet per second}}{\text{water area of each pass}}$$

$$= \frac{68{,}000 \times 3 \times 144}{62{\cdot}5 \times 60 \times 3500 \times 0{\cdot}336} \text{ ft. per second}$$

$$= 6{\cdot}66 \text{ ft. per second.}$$

At this velocity the tube friction for a 10-ft. tube is, from (fig. 7), 3·9 ft.. and for a 9-ft.-6-in. tube it will be

$$H_t = \frac{3{\cdot}9 \times 9{\cdot}5}{10}$$

$$= 3{\cdot}7 \text{ ft.}$$

From (fig. 7) $H_e = 1{\cdot}1$ ft., and for a three-flow condenser

$$H_c = 2 \text{ ft.}$$

The total loss in friction head then will be

$$\begin{aligned} H &= 3H_t + 3H_e + H_c \\ &= (3 \times 3{\cdot}7) + (3 \times 1{\cdot}1) + 2 \\ &= 11{\cdot}1 + 3{\cdot}3 + 2 \\ &= 16{\cdot}4 \text{ ft.} \end{aligned}$$

To reduce the friction loss some makers of condensing plant round off the entrance edge of the ferrules, thereby eliminating the vena contracta caused by the square edge of an ordinary ferrule.

The condition of the inside of the tube has a large effect on the friction head, and the rules given above are for reasonably clean tubes. As the tubes become coated with scale or muddy deposit the friction head rises, and it is common practice at power stations to fit gauges on the inlet and outlet cooling-water branches. When these gauges show a certain difference this is taken as an indication that the tubes need to be brushed or sponged out.

The great necessity for keeping the tubes clean in a modern condenser is not so much the increase of friction head caused by the dirty tubes, as the effect a slight deposit has on the heat flow from the steam to the water. This is readily realized, as the present day mean temperature difference runs so low as 12° or 13° F., and the increase of even 1° in this difference is a large percentage of the total.

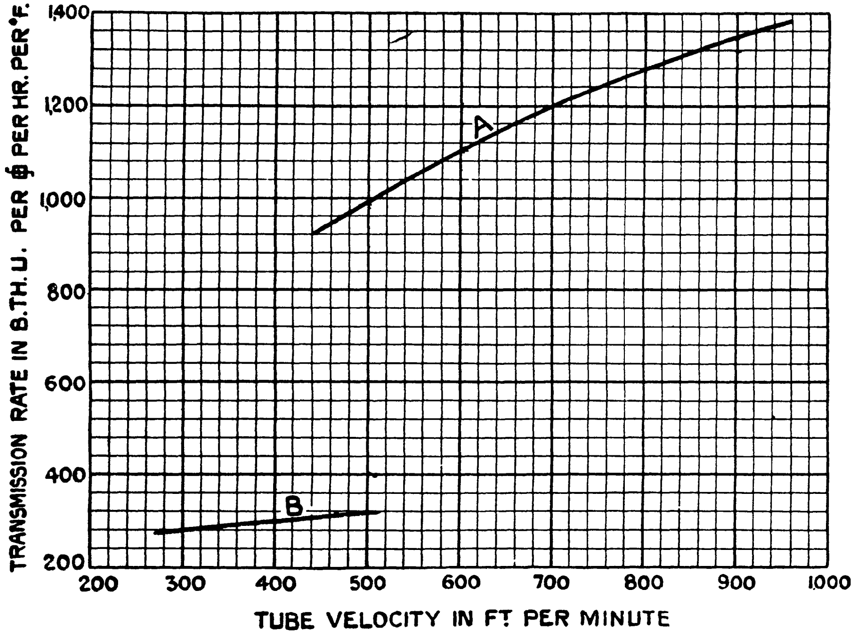


Fig. 8

TABLE I

Vacuum, Inches Hg.	Barometer, Inches Hg.	Vacuum Corrected to 30 in. Barometer, Inches Hg.	Condensate Temperature, Degrees F.	Inlet Temperature of Circulating Water, Degrees F.	B.					A.				
					Outlet Temperature, Degrees F.	Quantity of Water Passing, Lb./Min.	Tube Velocity, Ft./Min.	Transmission Rate, B.Th.U.	$\theta_{,,,}$ Degrees F.	Outlet Temperature, Degrees F.	Quantity of Water Passing, Lb./Min.	Tube Velocity, Ft. Min.	Transmission Rate, B.Th.U.	$\theta_{,,,}$ Degrees F.
28·25	29·44	28·81	90	53	57	74·6	513	307	35·0	62	137·0	942	1370	32·5
28·15	29·44	28·71	93	53	58	59·7	410	286	37·5	64	98·9	680	1135	34·5
28·1	29·44	28·66	94	53	59	51·6	354	293	38·0	66	83·0	570	1125	34·5
28·02	29·44	28·46	95	53	61	42·0	290	318	38·0	67	62·8	432	905	35·0
27·7	29·44	28·14	100	53	63	42·0	290	360	42·0	72	62·8	432	1145	37·5

The curves in fig. 8 show the results of an experiment on heat flow carried out by the author with two condenser tubes, one clean and one dirty. The dirty tube was taken out of the main condenser of the S.S. *City of Brisbane* after three voyages to Calcutta and back. These curves show the great resistance to heat flow of a thin deposit of scale, and the consequent necessity of keeping a condenser clean.

The heat flow is plotted as ordinates in B.Th.U. per hour per square foot of surface per degree Fahrenheit of mean temperature difference. This is known as the transmission rate. The tube velocity is plotted as abscissæ in feet per minute.

At a tube velocity of 500 ft. per minute the dirty tube allowed 320 B.Th.U. to pass, while the clean tube allowed 990 B.Th.U. to pass, nearly three times more.

The increase of friction head of the dirty tube over the clean tube is clearly shown by the smaller water quantity which the dirty tube allowed to pass. (See Table I.)

It is needless to state that when this condenser was cleaned the vacuum was greatly improved.

Owing to the difference in the conditions on which land and marine condensers have to operate, land condenser tubes have to be cleaned far more frequently than the tubes for marine condensers. The marine condenser has generally a greater tube velocity than the land condenser, and has not to operate with such muddy or oily water; the greater tube velocity tends to scour the tubes clean.

The deposit which is usually found in a marine condenser tube is a hard salt scale from the sea water, while that found in a land condenser tube is generally a soft deposit of mud and oil. This salt scale found in marine condenser tubes is not wholly due to the normal conditions of operation, as the cooling water temperature is far too low to produce the hard scale which is found; the scale is rather due to the heating up conditions of the main turbines. It is the practice with some marine engineers to heat up the main turbines with a low vacuum of 5 or 6 in. Hg in the condenser, this results in the condensers becoming very hot, so that the conditions become favourable for scaling.

TUBE CLEANING APPARATUS

Various devices have been invented to facilitate the cleaning of the tubes of surface condensers, and thereby reduce the losses due to the length of time the plant is shut down, as well as the expense associated with mechanical methods.

One of the most efficient devices for rapidly and thoroughly cleaning condenser tubes is the Mirrlees tube cleaning apparatus for surface condensers. This device is illustrated in fig. 9 and 10. The principle of operation is that a cleansing solution is caused to circulate through the tubes by the action

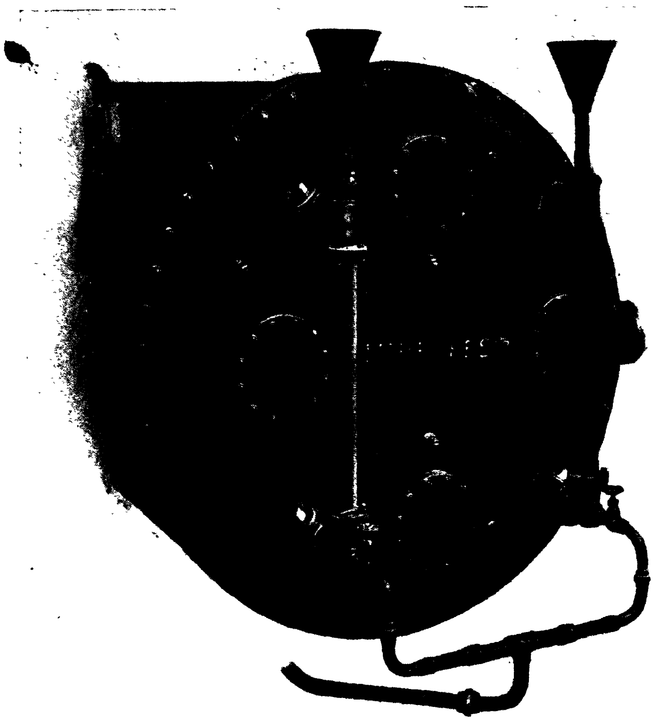


Fig. 9.—Surface Condenser Equipped with Tube Cleaning Apparatus—Duplex Type

of a steam jet ejector, which heats up the solution so that any scale, grease, or muddy deposit is quickly and thoroughly removed.

When the deposit is of a greasy or muddy nature the makers recommend a caustic-soda solution of 1 per cent, which may be gradually increased to 7 per cent, depending on the nature of the deposit.

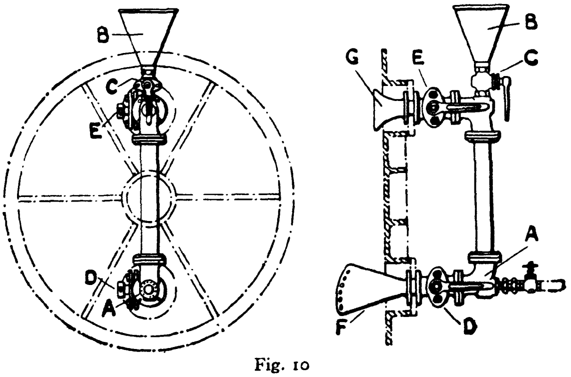


Fig. 10

When the deposit is in the form of a hard scale, a 5 per cent solution of hydrochloric acid is recommended, which may be increased in very bad cases to a 40 per cent solution.

Referring now to fig. 10, which shows the apparatus fitted to the end door of a surface condenser; A is the steam jet; B, a funnel for the introduction of the cleansing agent; C, D, and E are shut off cocks; G, a bell-mouth inlet orifice; and F is a perforated discharge cone.

The method of operation is as follows:

The inlet- and outlet-water valves on the main circulating system are closed, and the air cocks on the water boxes are opened. The condenser is then almost filled with a known quantity of clean water, and the air cocks closed. The cocks D and E are then opened and steam is turned on to the ejector A; this causes a circulation to take place through the tubes and water boxes; the water entering the orifice G, and passing down through the apparatus, is discharged through the discharge cone F by the action of the steam ejector. The water during the circulation is heated by the steam from the ejector. When the circulation is thus established, a known quantity of the cleansing agent is poured into the funnel B, this quantity being sufficient to give a solution of the required strength; the cock C is then opened and the cleansing agent is drawn into the apparatus, and mixed with the hot water circulating in the condenser.

The makers recommend that the cleaning operation should be continued for at least six to eight hours, and that the solution should be maintained at the proper strength by more cleansing agent being added from time to time; also, that the temperature of the solution in circulation should not be allowed to go over 150° F., but should be kept under this maximum by occasionally shutting down the ejector.

When the cleaning operation is finished the solution should be drained off or pumped out. The condenser should be thoroughly washed out with clean water when caustic soda has been used as a cleanser, and with an alkaline solution when hydrochloric acid has been used. Any acid which may be left is soon neutralized by the action of the scale-producing impurities in the circulating water.

CRITICAL VELOCITY IN TUBES

On p. 9 attention was drawn to the necessity of having the velocity over the critical, to ensure a rapid transference of heat from the tube wall to the cooling water, and it was there stated that under the critical velocity the water flows through the tubes in stream lines or in non-turbulent flow, and that over the critical velocity the flow is turbulent.

These two modes of flow were demonstrated by Professor Osborne Reynolds, and he also worked out formulæ so that the critical velocity could be determined for any given conditions. He further showed that at a velocity lower than the critical the law of friction head was

$$h \propto v,$$

where v is the tube velocity, and that at velocities over the critical the law was

$$h \propto v^n,$$

where $n = 1{\cdot}722$.

The previous rules given for the friction head in a condenser are for velocities over the critical.

The formula deduced by Reynolds from his experiments for the critical velocity is

$$v_c = \frac{1}{b} \times \frac{P}{d}, \quad \ldots\ldots\ldots\ldots\ldots\ldots (22)$$

where v_c is the critical velocity in feet per second;

b is the numerical constant $= 4{\cdot}06$;

P is Poiseuille's value for $\frac{\text{viscosity}}{\text{density}}$ of the water flowing through the tube;

d is the internal diameter of the tube in feet.

There are two critical velocities, a high velocity and a low velocity, and between these the motion is unstable. In condenser work it is only necessary to know the higher one, since it is essential that the velocity should never under any conditions fall below this. The above equation gives this higher value.

Now the value of P depends upon the temperature of the cooling water, and, since v_c is higher for cold water than it is for hot water, the inlet temperature of cooling water should be taken.

The value of P is given by the equation

$$P = \frac{1}{1 + 0{\cdot}0336T + 0{\cdot}000221T^2}, \quad \ldots\ldots\ldots\ldots (23)$$

where T is the inlet water temperature in degrees Centigrade.

The following table gives values of P for corresponding values of T for the usual temperatures met with in practice.

Values of T and P

T° F.	32	35	40	45	50	55	60	65	70	75	80
P	1	0·945	0·866	0·796	0·735	0·681	0·636	0·596	0·558	0·521	0·488

Should the reader desire more detailed information on this question of critical velocity he should consult a standard work on hydraulics.

The effect of this sudden change of mode of heat flow across a condenser tube can be very clearly demonstrated by a simple experiment with a condenser. If a thermometer placed in the outlet be carefully watched, and the tube velocity gradually increased, the temperature rise will at first be extremely slow, but immediately a certain velocity is reached a sudden jump of 4° or 5° will be seen on the thermometer.

Example.—Find the lowest permissible tube velocity in a surface condenser having $\frac{5}{8}$-in. O/D tubes, 18 S.W.G. thick, and having an inlet water temperature of 60° F.

From equation (22), p. 24.

$$v_c = \frac{1}{b} \times \frac{P}{d}.$$

$$d = \frac{0{\cdot}625 - 0{\cdot}048 \times 2}{12}$$

$$= \frac{0{\cdot}529}{12} \text{ ft.}$$

$$b = 4{\cdot}06,$$

and $$P = 0{\cdot}636.$$

$$\therefore v_c = \frac{12 \times 0{\cdot}636}{4{\cdot}06 \times 0{\cdot}529}$$

$$= 3{\cdot}55 \text{ ft. per second.}$$

This is the higher critical tube velocity, and is, therefore, the lowest permissible tube velocity for the efficient transference of heat.

The normal tube velocity in condensers is, of course, much higher than the critical velocity, but occasions do arise in which it is desirable to know what the critical velocity is. An example of this is to be found in the case of torpedo-boat destroyers. The condensers for these ships are designed with a very high tube velocity at full power, and a vacuum of about 27 in. to give as small and light a condenser as possible. For economy's sake these ships always cruise at reduced speeds, and the circulating pumps are therefore slowed down.

In the design of a condenser for a torpedo-boat destroyer the critical velocity should, therefore, be found and checked with the cruising conditions, so that the designed vacuum may be maintained without unduly cooling the condensate.

PRESSURE DROP OR VACUUM LOSS IN STEAM SPACE

In any steam condenser there is always a pressure drop on the steam side between the steam inlet and the air and vapour outlet. This drop is caused by the initiation of the velocity of the steam entering the condenser, and has a direct relation with the velocity and the density of the entering steam This drop is a direct loss in vacuum, and should be reduced to the minimum. It is becoming common practice for condenser makers to specify this drop for their condensers.

Let the following diagram, fig. 11, represent a condenser.

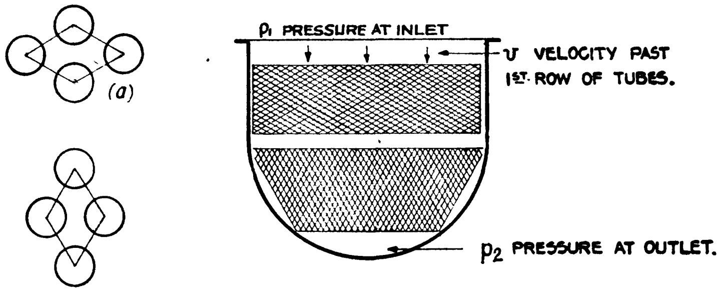


Fig. 11—Tube Spacing to minimize Pressure Drop

Let δ be the density of the entering steam in pounds per cubic foot;
v_s, the steam velocity past the first row of tubes in feet per second;
g, the acceleration due to gravity in feet per second per second;
p_1, absolute pressure at inlet;
p_2, absolute pressure at outlet;
d, the pressure drop in inches of mercury;
and h, the steam head in feet to initiate the velocity v_s.

Then
$$\delta \times h = p_1 - p_2 \text{ lb. per square foot}$$
$$= 144(P_1 - P_2) \text{ lb. per square inch.}$$

Now
$$h = \frac{v_s^2}{2g}.$$
$$\therefore \delta \times \frac{v_s^2}{2g} = 144(P_1 - P_2) \text{ lb. per square inch.}$$

Again
$$\frac{P_1 - P_2}{14{\cdot}7} = \frac{d}{29{\cdot}992}.$$

By substitution
$$\delta \times \frac{v_s^2}{2g} = \frac{d \times 14{\cdot}7 \times 144}{29{\cdot}992},$$

or
$$v_s = \sqrt{\frac{2g \times 14{\cdot}7 \times 144}{29{\cdot}992}} \times \sqrt{\frac{d}{\delta}}$$
$$v_s = 67{\cdot}2\sqrt{\frac{d}{\delta}}.$$

The above expression must be multiplied by a factor K, which depends on the depth of the penetration of the steam into the tube banks. This factor K will naturally vary with the design of and the method of tube spacing in the condenser, and varies from 0·2 to 0·5. A good average value which agrees well with actual test figures is 0·314.

Taking this value for K we get

$$v_s = 67{\cdot}2\mathrm{K}\sqrt{\frac{d}{\delta}}$$

$$= 0{\cdot}314 \times 67{\cdot}2\sqrt{\frac{d}{\delta}}$$

$$= 21{\cdot}1\sqrt{\frac{d}{\delta}} \qquad (24)$$

This equation enables us to estimate the pressure drop from a known entering steam velocity and vacuum. The density δ is the density corresponding to the vacuum at the top of the condenser, and can be found from the steam tables.

For given pressure drops the author has plotted the curves given in fig. 12, and these agree approximately with values obtained from actual condensers.

These curves show that for the same pressure drop a much higher entering steam velocity can be allowed for high vacua than for low vacua; this, of course, is due to the great decrease in density at the high vacua.

When giving guarantees of a condenser it is usual to give performance curves at varying vacua and inlet water temperatures, and in any design care should be taken to check the pressure drop at the highest guaranteed vacuum, because for a constant steam quantity the greatest drop obtains at the highest vacuum.

This can be proved from equation (24).

Thus $$v_s = 21{\cdot}1\sqrt{\frac{d}{\delta}},$$

or $$d = \delta\left(\frac{v}{21{\cdot}1}\right)^2 \qquad (25)$$

Now if the steam quantity Q be constant and the entrance area A is constant, we have

$$v_s = \frac{\mathrm{Q}}{\mathrm{A} \times \delta}.$$

Then, substituting this value of v in (25), we get

$$d = \delta\left(\frac{\mathrm{Q}}{21{\cdot}1\mathrm{A}\delta}\right)^2$$

$$= \frac{1}{\delta}\left(\frac{\mathrm{Q}}{21{\cdot}1\mathrm{A}}\right)^2$$

$$= \text{constant} \times \frac{1}{\delta}.$$

The drop therefore varies inversely as the density, so that the maximum drop is obtained at the minimum density, that is, at the highest vacuum.

To keep down the pressure drop the long axis of standard 60° tube pitching should be placed horizontally, as in this position the entrance area for the steam is greater than when the long axis is placed vertically;

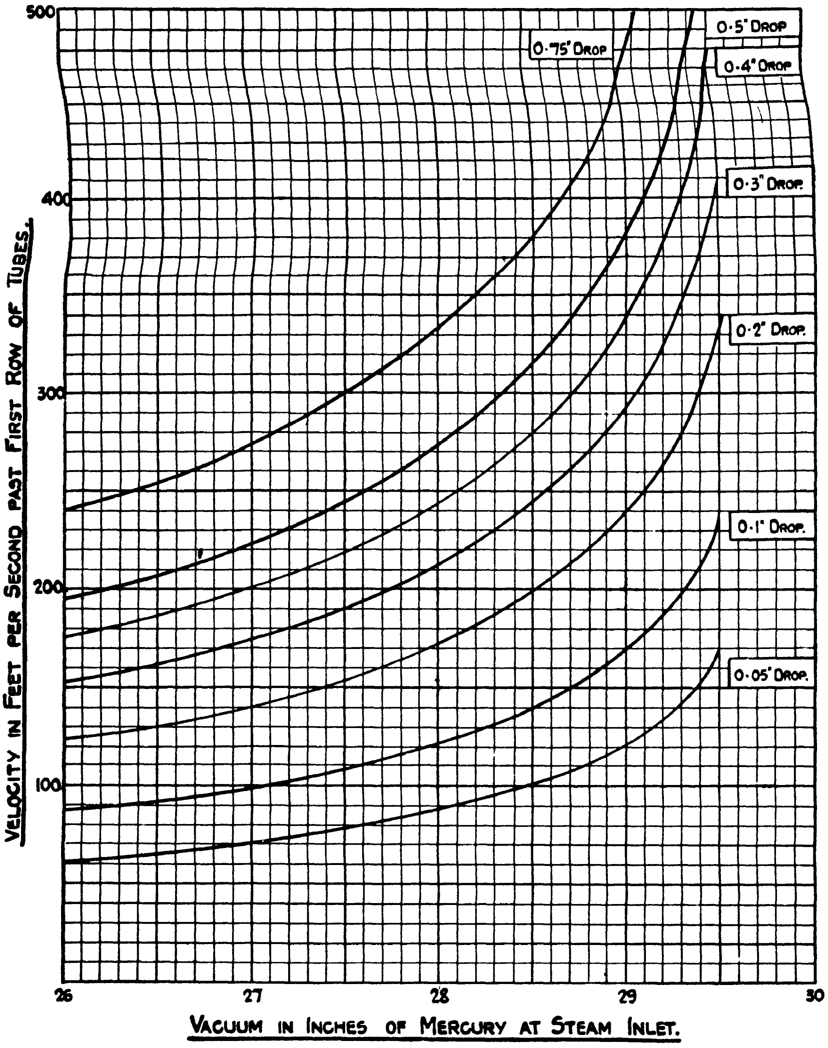


Fig. 12.—Relation between Pressure Drop, Vacuum, and Steam Velocity past First Row of Tubes

that is, the pitching should be placed as shown at (*a*) fig. 11. This helps to keep the condenser a reasonable breadth. This is particularly convenient with the modern underhung condenser, where the available breadth is limited by the foundations or seatings of the main turbine.

VARIATION IN THE RATE OF CONDENSATION THROUGHOUT A SURFACE CONDENSER

For the purpose of this investigation take a two-flow condenser as shown in fig. 13.

The temperatures and vacua stated above are those obtained from an actual condenser.

Consider the points A, B, C, and D. The temperature differences at the four points are:

at point A, $79 - 67 = 12^\circ$ F.
" B, $79 - 58 = 21^\circ$ F.
" C, $71 - 58 = 13^\circ$ F.
" D, $71 - 50 = 21^\circ$ F.

The condensation rates at the same four points vary according to equation (13), p. 9

$$w = \frac{\theta_m}{R_a + R_t + R_w}$$

Now R_a varies considerably from the top to the bottom, depending on the air richness of the steam. At the top the percentage of air to steam is very small, and the value may be taken as that given on p. 10, namely, 0·48; while at the bottom the percentage is very high, and the value of R_a may go as high as 4. The values given on p. 10 are average working values for the whole condenser.

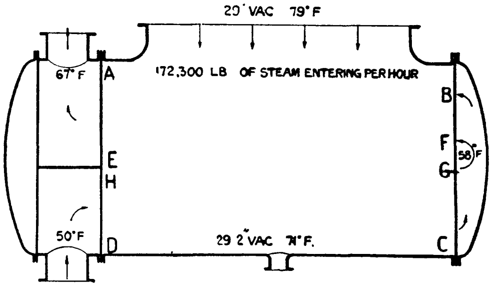


Fig. 13

The condenser under consideration had a very efficient air pump, so that the condensate temperature was maintained close to the temperature corresponding to the vacuum. In this case, therefore, the value of R_a may be taken as 1·4.

At the points A, B, C, and D the values of w become:

$$\text{at point A} \quad w = \frac{12}{0{\cdot}48 + 0{\cdot}08 + R_w},$$

$$\text{and} \quad R_w = \frac{C}{v^{0\cdot82}} \quad \text{(equation 14, p. 9; the outside diameter of tube is } \tfrac{3}{4} \text{ in.)}$$

$$= \frac{3{\cdot}05}{6^{0\cdot82}}$$

$$= 0{\cdot}7.$$

$$\therefore w = \frac{12}{1{\cdot}26}$$

$$= 9{\cdot}5 \text{ lb. per square foot.}$$

Similarly at B, $w = \frac{21}{1 \cdot 22} = 17 \cdot 2$ lb. per square foot.

" C, $w = \frac{13}{2 \cdot 24} = 5 \cdot 8$ lb. per square foot.

" D, $w = \frac{21}{1 \cdot 4 + 0 \cdot 08 + 0 \cdot 91}$

$= 8 \cdot 8$ lb. per square foot.

If the above process be applied to the points E, F, G, and H, the points E and F being on the top flow and the points G and H being on the bottom flow, it will be found that there is a big difference between *w* at E and *w* at H, and that *w* at F and G will be the same.

The author has worked these out and plotted, in fig. 14, the values of *w* at the points marked. The values at intermediate points are indicated roughly by the curves. These show the great variation in condensation rate which takes place in a surface condenser. The full line curve is for the water-box end, and the dotted line curve is for the return end. The maximum variation is from 17·2 to 3·17, or a variation of over 400 per cent.

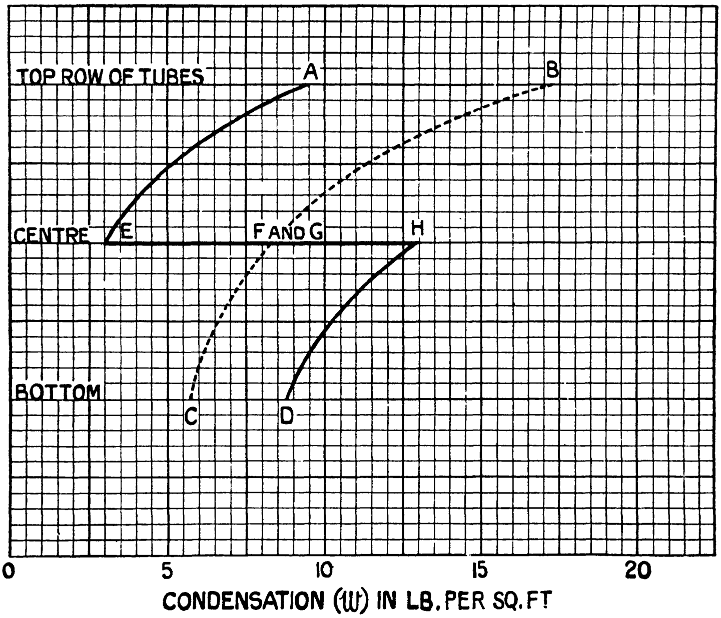


Fig. 14.—Variation in Rate of Condensation throughout Surface Condenser

In all the rules given above, *w* signifies the average condensation rate throughout the condenser.

SHAPE OF CONDENSERS FOR CONSTANT STEAM VELOCITY

Many condensers are designed to give an approximately constant velocity to the steam during its passage through the condenser; that is, the flow area is reduced as the steam condenses.

The main reason for maintaining the steam velocity is to ensure that the tubes at the bottom are steam swept as well as those at the top, so that the air blanket, which collects as the steam condenses, is brushed off the tubes and the whole surface is rendered effective for the condensation of steam, which, of course, is the function of a steam condenser. A well-designed condenser should have no stagnant zones wherein air may collect.

Probably the clearest method of showing the proper shape of a condenser is to work out a concrete example.

Let us take the condenser shown in fig. 13. This condenser is to condense 172,300 lb. of steam per hour at a vacuum of 29 in. Hg, with cooling water at the inlet of 50° F. The temperature rise in the cooling water is 17° F., so that the mean temperature difference will be

$$\begin{aligned}\theta_m &= \frac{\theta_1 - \theta_2}{2{\cdot}3 \log_{10}\frac{\theta_1}{\theta_2}} \\ &= \frac{21 - 12}{2{\cdot}3 \times \log_{10} 1{\cdot}75} \\ &= \frac{9}{2{\cdot}3 \times 0{\cdot}243} \\ &= 16{\cdot}1^\circ \text{ F.}\end{aligned}$$

Taking $R_a = 1{\cdot}05$ and $v = 6$ ft. per second and $\frac{3}{4}$-in. O/D tubes, and

$$\begin{aligned}R_w &= \frac{C}{v^{0{\cdot}82}} \\ &= \frac{3{\cdot}48}{6^{0{\cdot}82}} \\ &= 0{\cdot}80;\end{aligned}$$

then

$$\begin{aligned}w &= \frac{16{\cdot}1}{1{\cdot}05 + 0{\cdot}08 + 0{\cdot}80} \\ &= \frac{16{\cdot}1}{1{\cdot}93} \\ &= 8{\cdot}34 \text{ lb. per square foot.}\end{aligned}$$

The total condensing surface therefore will be

$$\begin{aligned}S &= \frac{172{,}300}{8{\cdot}34} \\ &= 20{,}700 \text{ sq. ft.}\end{aligned}$$

Again, assuming that each pound of steam entering the condenser has 1000 B.Th.U. extracted from it in condensing, we get the total quantity of cooling water flowing through the tubes from equation (17) to be

$$G = \frac{10W}{6(T_2 - T_1)}$$
$$= \frac{10 \times 172{,}300}{6 \times 17}$$
$$= 16{,}900 \text{ gall. per minute.}$$

Let n be number of tubes per flow, and A the cross-sectional area of the tube in square inches, then

$$\frac{n \times A \times v}{144} = \frac{G}{60 \times 6{\cdot}25},$$

i.e. $$n = \frac{16{,}900 \times 144}{0{\cdot}336 \times 6 \times 60 \times 6{\cdot}25}$$
$$= 3220 \text{ tubes.}$$

To find the length l between the tube plates we have:

$$S = \frac{2\pi D l n}{12},$$

i.e. $$l = \frac{12S}{2\pi D n},$$

where S is the tube surface and D the outside diameter of the tube in inches,

i.e. $$l = \frac{12 \times 20{,}700 \times 4}{\pi \times 3 \times 2 \times 3220}$$
$$= 16{\cdot}4 \text{ ft.}$$

We have thus all the condenser particulars except the number of tubes in each row. It is this number which determines the shape of the condenser.

Let P be the pitch of the tubes in inches;
D, the external diameter of the tubes in inches;
N, the number of tubes per row;
C_0, the total condensation;
W_1, W_2, &c., the condensation in 1st, 2nd, &c., rows respectively, counting from the steam inlet;
S_1, S_2, &c., the tube surface in 1st, 2nd, &c., rows respectively;
a, the volume in cubic feet passing any row;
V, the specific volume in cubic feet of the steam at any row;
v_s, the steam velocity in feet per second; and
w_1, w_2, &c., the condensation per square foot at the 1st, 2nd, &c., rows, respectively.

Now the volume *a* of steam which passes the *n*th row per second

$$= \left[\frac{(P - D)(N_n + 1)}{12}\right] l;$$

but, since the volume of steam passing the *n*th row per second must be equal to the total volume of steam condensed per second below the *n*th row, this volume

$$= \frac{\left[C_0 - \left\{\frac{W_1}{2} + \left(\frac{W_1}{2} + \frac{W_2}{2}\right) + \ldots + \left(\frac{W_{n-1}}{2} + \frac{W_n}{2}\right)\right\}\right] V}{3600 v_s},$$

$$\therefore \left[\frac{(P - D)(N_n + 1)}{12}\right] l = \frac{\left[C_0 - \left\{\frac{W_1}{2} + \left(\frac{W_1}{2} + \frac{W_2}{2}\right) + \ldots + \left(\frac{W_{n-1}}{2} + \frac{W_n}{2}\right)\right\}\right] V}{3600 v_s}.$$

Now $W_1 = w_1 S_1$, and $W_2 = w_2 S_2$, &c.

$$\therefore \left[\frac{(P - D)(N_n + 1)}{12}\right] l$$

$$= \frac{\left[C_0 - \left\{\frac{w_1 S_1}{2} + \left(\frac{w_1 S_1}{2} + \frac{w_2 S_2}{2}\right) + \ldots + \left(\frac{w_{n-1} S_{n-1}}{2} + \frac{w_n S_n}{2}\right)\right\}\right] V}{3600 v_s} \quad . (26)$$

From this general equation the number of tubes in each row can be determined. The number of tubes in the first row is given by

$$\left[\frac{(P - D)(N_1 + 1)}{12}\right] l = \frac{\left[C_0 - \frac{w_1 S_1}{2}\right] V}{3600 v_s};$$

for the second row:

$$\left[\frac{(P - D)(N_2 + 1)}{12}\right] l = \frac{\left[C_0 - \left\{\frac{w_1 S_1}{2} + \left(\frac{w_1 S_1}{2} + \frac{w_2 S_2}{2}\right)\right\}\right] V}{3600 v_s};$$

and so on for each row.

The values of w_1, w_2, &c., have to be found independently from the θ_m at the row and the value of R_a.

As was stated on pp. 11 and 29, R_a varies considerably from top to bottom of the condenser, and may be taken as 0·48 for the top row and progressively increased to 3 for the bottom few rows.

The above is the general equation, and for any specific condenser it simplifies the calculation to gather the fixed values into a constant thus:

$$S_1 = \frac{\pi D l N_1}{12}$$

$$= \frac{\pi \times 3 \times 16{\cdot}4 \times N_1}{4 \times 12}$$

$$= 3{\cdot}22 N_1,$$

$$\text{and} \qquad S_n = 3{\cdot}22 N_n.$$

We have, therefore, by substitution in equation (26) and taking $P = 1\frac{1}{4}$ in., $D = \frac{3}{4}$ in., and $v_s = 200$ ft. per second, which velocity gives a reasonable pressure drop (see fig. 12):

$$\frac{(1\cdot25-0\cdot75)(N_n+1)\,16\cdot4}{12} = \frac{[C_0-\{w_1N_1+(w_1N_1+w_2N_2)+\ldots+(w_{n-1}N_{n-1}+w_nN_n)\}3\cdot22/2]V}{3600\times200},$$

$$\text{i.e. } N_n = \frac{[C_0-\{w_1N_1+(w_1N_1+w_2N_2)+\ldots+(w_{n-1}N_{n-1}+w_nN_n)\}1\cdot61]V}{490{,}000}-1 \ldots (27)$$

For the first row, where the vacuum is 29 in. and $V = 657$, we have:

$$\theta_m = 16\cdot1^\circ \text{ F. (see p. 31)},$$

$$\text{and} \quad w_1 = \frac{\theta_m}{R_a + R_t + R_w},$$

$$\text{where} \quad R_a = 0\cdot48,$$

$$\text{and} \quad R_w = \frac{C}{(\text{tube velocity})^{0\cdot82}}.$$

C, for $\frac{3}{4}$-in. tubes and a mean temperature of $\left(\frac{58+67}{2}\right)$, $= 62\cdot5$ is, from fig. 5, 3·27; and the tube velocity was stated on p. 31 as 6 ft. per second.

$$\therefore R_w = \frac{3\cdot27}{6^{0\cdot82}}$$

$$= \frac{3\cdot27}{4\cdot34}$$

$$= 0\cdot753.$$

$$\text{Then} \quad w_1 = \frac{16\cdot1}{0\cdot48 + 0\cdot08 + 0\cdot753}$$

$$= 12\cdot25 \text{ lb. per square foot.}$$

$\therefore$ by substitution in equation (27) we get

$$N_1 = \frac{[172{,}300 - 12\cdot25N_1 \times 1\cdot61]657}{490{,}000} - 1$$

$$= 231 - 0\cdot026N_1 - 1$$

$$= \frac{230}{1\cdot026}$$

$$= 224 \text{ tubes in first row.}$$

For the second row θ_m will be substantially the same, and, putting $R_a = 0\cdot5$, we get

$$w_2 = \frac{16\cdot1}{0\cdot5 + 0\cdot08 + 0\cdot753}$$

$$= 12\cdot08 \text{ lb. per square foot.}$$

$$\text{Then } N_2 = \frac{[172{,}300 - \{12{\cdot}25 \times 224 + 12{\cdot}25 \times 224 + 12{\cdot}05N_2\}1{\cdot}61]657}{490{,}000} - 1$$
$$= 231 - 11{\cdot}85 - 0{\cdot}026N_2 - 1$$
$$= 213 \text{ tubes in second row.}$$

For the third row θ_m will again be substantially the same, and, putting $R_a = 0{\cdot}52$, we get

$$w_3 = \frac{16{\cdot}1}{0{\cdot}52 + 0{\cdot}08 + 0{\cdot}753}$$
$$= 11{\cdot}9 \text{ lb. per square foot.}$$

Then

$$N_3 = 231 - 11{\cdot}85 - \frac{[\{12{\cdot}05N_2 + 12{\cdot}05N_2 + 11{\cdot}85N_3\}1{\cdot}61]657}{490{,}000} - 1$$
$$= 231 - 11{\cdot}85 - 11{\cdot}08 - 0{\cdot}256N_3 - 1$$
$$= \frac{207}{1{\cdot}026}$$
$$= 202 \text{ tubes in third row.}$$
$$N_4 = 231 - 11{\cdot}85 - 11{\cdot}08 - \frac{[\{w_3N_3 + w_3N_3 + w_4N_4\}1{\cdot}61]657}{490{,}000} - 1.$$

By continuing this process from top to bottom of the condenser the number of tubes in each row can be found This the author has done, and the result is shown in fig. 15. This is the theoretically correct shape at the mid section for constant steam velocity, and a constant pitch of tubes. The sections at the ends would be different from the mid section due to the difference of mean temperature difference. Such a condenser would, of course, be impossible, as the tubes could not be accommodated at the narrow end section.

The extreme breadth of the steam entrance should be noted, and the problem of allowing sufficient area between the tubes for the entering steam is one of the difficulties of good modern condenser design. This difficulty has become very real recently, due to the present-day search after economy and the reduction or elimination of all losses.

As stated on p. 25, the pressure drop is a direct loss and should be reduced to a minimum, so that the steam velocity must be kept down by allowing sufficient entrance area; this is not always easily accomplished because of the limited space under a modern turbine.

The difficulty of the very wide entrance can be got over by varying the pitch of the tubes from top to bottom. That is, the tubes can be widely spaced at the top, and the pitch reduced in steps till normal pitching is reached. Such a condenser is shown in fig. 16.

Some condenser makers get over the difficulty by arranging lanes in the top-tube nest, so that the steam velocity past the first row is reasonable without the condenser being unduly broad.

In all modern condensers it will generally be found that the highest steam velocity is that past the first row, or in other words the steam is throttled through the first row, and in consequence of this the pressure drop is mainly in the top nest of tubes; if the experiment be tried of fitting three mercury columns to the steam space of a condenser—one at the top, one half-way down, and one at the bottom—it will be found that quite four-fifths of the total pressure drop or loss of vacuum is across the top nest. This is because of the above throttling action through the first row of tubes. For practical work it is therefore generally sufficient to ensure that the steam velocity past the first row of tubes is not excessive, and to neglect the velocity at any lower row.

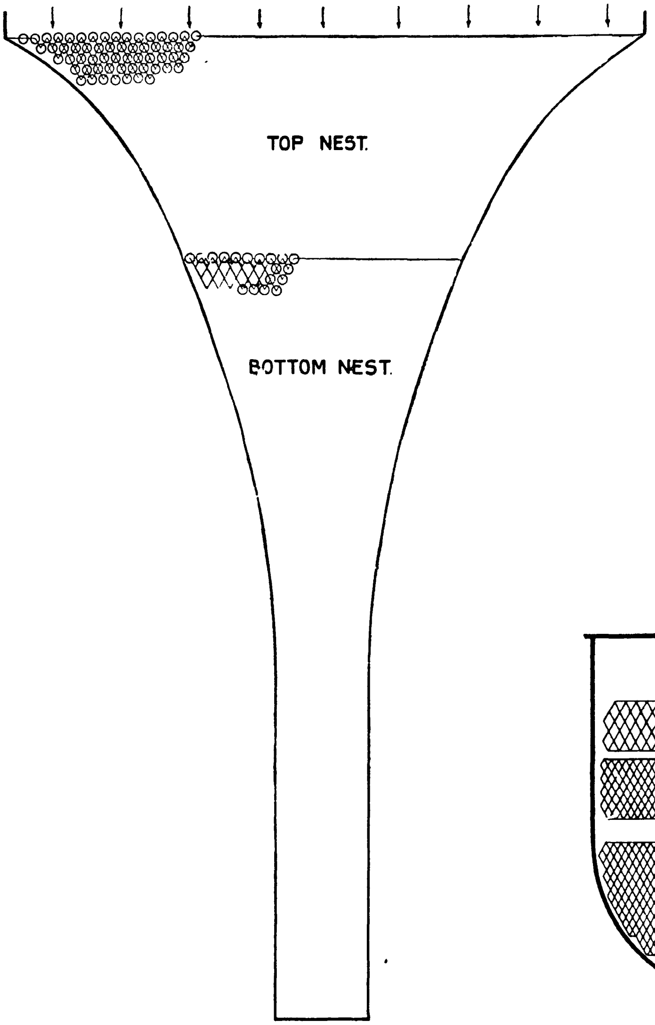


Fig. 15

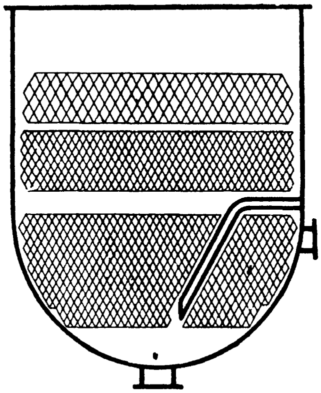


Fig. 16

MASS OF AIR AND VAPOUR TO BE WITHDRAWN FROM A CONDENSER TO MAINTAIN A GIVEN VACUUM

The mass of air and vapour to be withdrawn from a condenser depends upon two factors:

1. The mass of air which leaks into the vacuum system.
2. The mass of air which comes over with the steam.

The first of these factors depends mainly upon workmanship, and can, with care taken in the design and making of the vacuum joints, be reduced to a small figure.

The second depends upon the treatment the feed water receives before it enters the boiler. This air enters the boiler as entrained air in the feed water and is a relatively small quantity. It is usual to make an overall air allowance which is the sum of the above two quantities.

The British Electrical and Allied Manufactures Association have drawn up a standard for land surface condensing plants, and the rule given by them is as follows:

If W is the total weight of steam condensed in pounds per hour,
and m the mass of air to be withdrawn in pounds per hour,
then $m = \frac{0{\cdot}5\text{W}}{1000} + 3$ lb. per hour.

This air quantity is not to be regarded as the actual air leakage in the vacuum system, but is only an allowance for general work. The author has found that it is generally possible in an actual plant to reduce this leakage to about half the above quantity. In this connection the author has found large differences with various makes of turbine; this can be accounted for by the designs of the L.P. gland of the different makers.

Some designs seem to be quite leak proof, while others need to have a good blow of steam coming from the gland to seal them effectively. The water sealed gland is quite effective, and is generally only found in land turbines. Marine turbines have steam sealed glands which are quite effective.

Let m_1 be the mass of air leaking into the vacuum system in pounds per hour;
m_2, the mass of air coming over with the steam in pounds per hour;
then $m = m_1 + m_2$.

Let t be the temperature of air and vapour leaving the condenser, degrees F.;
p_s, the vapour pressure in inches of mercury absolute;
p_a, the air pressure in inches of mercury absolute; and
p, the total absolute pressure of air and vapour in inches of mercury; then, by Dalton's law of partial pressures, we have

$$p = p_a + p_s.$$

Now, as was explained on pp. 12–14, the partial air pressure p_a can always be found if t is known; then knowing p_a we can, by Charles's law, find the specific volume of the air at temperature t.

We know that 1 lb. of dry air at 32° F. occupies a volume of 12·39 c. ft., and by Charles's law

$$\frac{\mathrm{PV}}{\mathrm{T}} = \text{constant}, \therefore \frac{30 \text{ in.} \times 12{\cdot}39}{459{\cdot}6 + 32} = \frac{p_a \mathrm{V}_a}{459{\cdot}6 + t},$$

$$\text{or} \quad \mathrm{V}_a = \frac{30 \times 12{\cdot}39(459{\cdot}6 + t)}{p_a(459{\cdot}6 + 32)}. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (28)$$

Knowing then the partial air pressure p_a, the specific volume V_a can be found from the above expression.

Again, we know, from the theory of a mixture of gases, that each gas or vapour occupies the same volume and exerts its own pressure, so that the vapour which is associated with this air will occupy a volume V_a and exert its own pressure p_s.

Now from the steam tables we find that the specific volume of the vapour at pressure p_s is V_s, so that the mass of vapour which occupies a volume V_a will be

$$\frac{\mathrm{V}_a}{\mathrm{V}_s} \text{ lb.}$$

Again, since 1 lb. of air occupies the volume V_a, and $\frac{\mathrm{V}_a}{\mathrm{V}_s}$ lb. of vapour occupies the same volume, the total mass of mixture occupying the volume V_a will be $1 + \frac{\mathrm{V}_a}{\mathrm{V}_s}$ lb., so that the total mass M of mixture to be withdrawn from the condenser per hour to maintain the absolute pressure p is

$$\mathrm{M} = m\left(1 + \frac{\mathrm{V}_a}{\mathrm{V}_s}\right). \quad \ldots\ldots\ldots\ldots\ldots\ldots (29)$$

Example.—A surface condenser is to condense 202,000 lb. of steam per hour and is to maintain a vacuum at the bottom of the condenser of 28·25 in. mercury. The temperature of the air and vapour leaving the condenser is estimated to be 82° F. Find the mass of air and vapour the air pump must be designed to deal with. The dry air allowance is

$$\begin{aligned} m &= \frac{0{\cdot}5\mathrm{W}}{1000} + 3 \text{ lb. per hour.} \\ &= \frac{0{\cdot}5 \times 202{,}000}{1000} + 3 \\ &= 104 \text{ lb. per hour.} \end{aligned}$$

From the steam tables we find that $p_s = 1{\cdot}1$ in., so that

$$\begin{aligned} p_a &= p - p_s \\ &= (30 - 28{\cdot}25) - 1{\cdot}1 \\ &= 0{\cdot}65 \text{ in. of mercury absolute.} \end{aligned}$$

Now $$V_a = \frac{30 \times 12{\cdot}39(459{\cdot}6 + 82)}{0{\cdot}65(459{\cdot}6 + 32)}$$
$$= \frac{30 \times 12{\cdot}39 \times 541{\cdot}6}{0{\cdot}65 \times 491{\cdot}6}$$
$$= 630 \text{ c. ft. per pound of air.}$$

Again, from the steam tables we find that the specific volume of vapour at 82° F. is 598·7, i.e. $V_s = 598{\cdot}7$ c. ft. per pound.

$$\therefore M = m\left(1 + \frac{V_a}{V_s}\right)$$
$$= 104\left(1 + \frac{630}{598{\cdot}7}\right)$$
$$= 104 \times 2{\cdot}052$$
$$= 214 \text{ lb. per hour.}$$

Example.—Find the mass of vapour which is associated with 1 lb. of air at a temperature of 70° F. and an absolute pressure of 0·79 in. of mercury.

From the steam tables $p_s = 0{\cdot}74$ in. of mercury.

Now $$p_a = p - p_s$$
$$= 0{\cdot}79 - 0{\cdot}74$$
$$= 0{\cdot}05 \text{ in. of mercury.}$$
$$V_a = \frac{30 \times 12{\cdot}39(459{\cdot}6 + 70)}{0{\cdot}05(459{\cdot}6 + 32)}$$
$$= \frac{30 \times 12{\cdot}39 \times 529{\cdot}6}{0{\cdot}05 \times 491{\cdot}6}$$
$$= 8000 \text{ c. ft. per pound of air.}$$

Again, from the steam tables $V_s = 871$ c. ft. per pound.

$$\therefore M = m\left(1 + \frac{V_a}{V_s}\right)$$
$$= 1\left(1 + \frac{8000}{871}\right)$$
$$= 1 + 9{\cdot}19$$
$$= 10{\cdot}19 \text{ lb. per pound of air.}$$

To facilitate calculations of this sort the author has prepared the graphs shown on fig. 17. A study of these graphs shows the rapid increase in the weight of mixture to be removed as the temperature approaches that corresponding to the vacuum; when the temperature and vacuum are corresponding the curves go to infinity, so that, unless a cooling compartment be fitted to the condenser, it is impossible to maintain a vacuum corresponding to the condensate temperature if there be the smallest

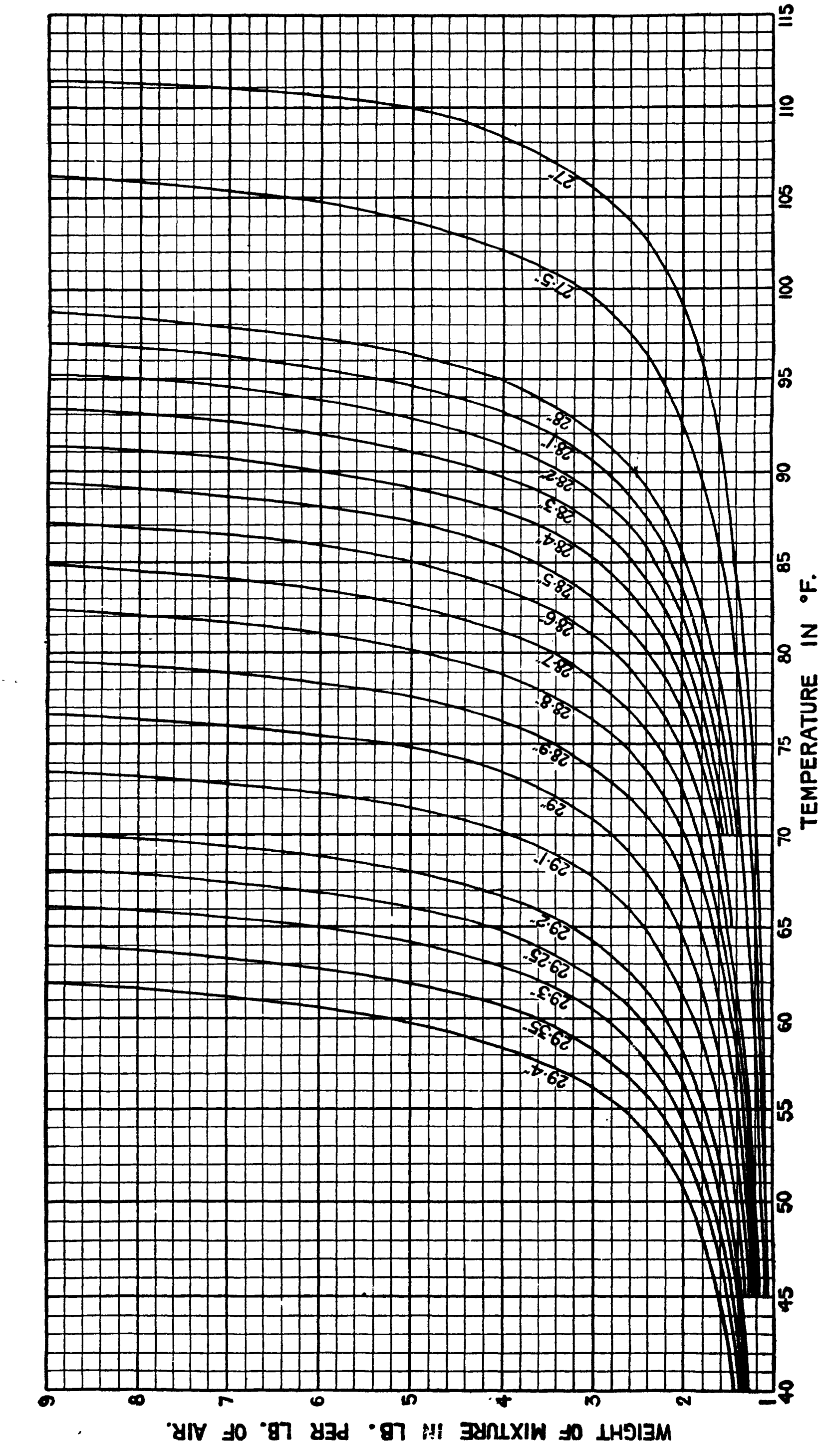


Fig. 17.—Relation between Weight, Vacuum, and Temperature of Air and Vapour

quantity of air present, as inevitably there must be. For this reason and to reduce the size of the air pumping plant, modern condensers for turbines are always fitted with an air cooler.

AIR COOLER FITTED TO SURFACE CONDENSER

The air cooler fitted to the modern surface condenser for turbines is usually formed integral with the condenser itself by fitting a baffle plate over a portion of the tube surface, and taking the dry-suction branch from the top of the cooler thus formed. When fitting such a cooler care should be taken that a good joint is formed between the baffle plate, tube plates, and the tube-supporting or diaphragm plates, as otherwise the air-cooling effect will be destroyed by leakage. To get the maximum air-cooling effect the air and vapour must pass under the lower edge of the baffle plate, and flow upwards through the cooler to the dry-suction outlet. The dry-suction branch should not be fitted immediately under the top of the baffle plate, but two or three rows of tubes should be arranged between the top of the baffle and the dry-suction branch. This is to minimize the heating effect of the hot vapour, on the condensing side of the baffle, on the air which has already been cooled in the cooler. In other words, the air which has been cooled should be protected from being reheated by coming into contact with the relatively hot baffle plate. A dry-suction branch should be fitted between each diaphragm plate and the branches connected externally by a breeches pipe, or alternatively, a few tubes can be left out opposite each branch and a hole cut in each diaphragm plate where the tubes have been left out. By this means each cooler compartment does its share of the work of air cooling. A good example of a modern integral air cooler is shown in fig. 16.

Some recent condensers built in the United States have been fitted with external air coolers; these coolers have cast-iron tubes with fins cast on to give a large cooling surface in contact with the air.

The integral air cooler has the advantage that no special cold-water connections are necessary, and generally gives a much neater arrangement.

Let fig. 18 represent an integral air cooler.

Let p_1 be the absolute pressure at inlet;
t_1, the temperature at inlet;
p_2, the absolute pressure at outlet; and
t_2, the temperature at outlet.

Then the partial air pressure at inlet will be

$$p_{1a} = p_1 - p_{1s}, \quad \dots\dots\dots\dots\dots\dots (30)$$

and at outlet it will be

$$p_{2a} = p_2 - p_{2s}. \quad \dots\dots\dots\dots\dots\dots (31)$$

If V_{1a} be the air volume at inlet, V_{2a} the air volume at outlet, and V_{1s}

and V_{2s} the specific volume at inlet and outlet respectively, then the mass of vapour condensed out by the cooler is:

$$M_s = \left(\frac{V_{1a}}{V_{1s}} - \frac{V_{2a}}{V_{2s}}\right)m. \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(32)$$

Here m is, as before, the mass of the air, V_{1a} and V_{2a} can be found from equation (28), when p_{1a} and p_{2a} are found from equations (30) and (31), and V_{1s} and V_{2s} can be read off the steam tables.

Now the heat given up by the condensation of this vapour plus the heat given up by the cooling of the air is the total heat transference across the cooling surface of the tubes.

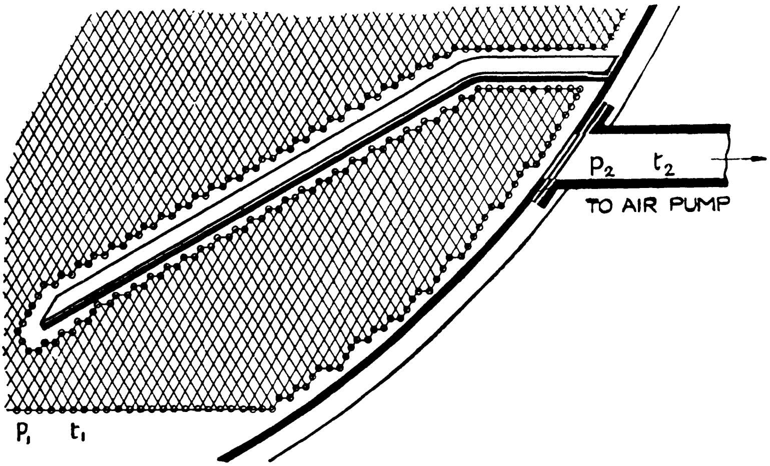


Fig. 18.—Section of Air Cooler for Surface Condenser

If L be the latent heat of the vapour at pressure p_{1s}, and γ the specific heat of the air $= 0{\cdot}2375$, we get that the heat transference H across the tube surface in B.Th.U. per hour is

$$\begin{aligned} H &= M_sL + m\gamma(t_2 - t_1) \\ &= m\left[L\left\{\frac{V_{1a}}{V_{1s}} - \frac{V_{2a}}{V_{2s}}\right\} + \gamma(t_2 - t_1)\right]. \qquad \ldots\ldots\ldots\ldots(33) \end{aligned}$$

As already proved (p. 16), the heat taken up by the cooling water is

$$H = 600G(T_2 - T_1).$$

But this quantity of heat is the heat transference across the tubes.

$$\therefore 600G(T_2 - T_1) = m\left[L\left\{\frac{V_{1a}}{V_{1s}} - \frac{V_{2a}}{V_{2s}}\right\} + \gamma(t_1 - t_2)\right]. \quad \ldots\ldots(34)$$

From this equation the outlet temperature of the cooling water T_2 can be found, since all the other quantities are known, and knowing T_2 we can find the mean temperature difference θ_m in the cooler.

θ_m will then be

$$\theta_m = \frac{(t_1 - T_1) - (t_2 - T_2)}{2{\cdot}3 \times \log_{10}\frac{(t_1 - T_1)}{(t_2 - T_2)}}. \quad \text{(See p. 45.)}$$

Again, from equation (13) (p. 9) we have

$$w = \frac{\theta_m}{R_w + R_t + R_f + R_a},$$

or summing R_f and R_a as before and calling the sum R_a we have

$$w = \frac{\theta_m}{R_w + R_t + R_a}.$$

Knowing the values of the resistances R_w, R_t, and R_a for cooler conditions, we can therefore determine the condensation per square foot of surface, and since the total condensation M_s can be determined from equation (32) the condensing surface S_c can be found, i.e.

$$S_c = \frac{M_s}{w} = \frac{m\left(\frac{V_{1a}}{V_{1s}} - \frac{V_{2a}}{V_{2s}}\right)(R_w + R_t + R_a)}{\theta_m} \quad \ldots\ldots\ldots\ldots(35)$$

For cooler conditions the practical working values of the resistances are:

R_a = 3 to 4,
R_t = 0·08 (as before),
R_w is found from fig. 5 (as before).

Example.—A surface condenser is supplied with cooling water at 80° F.; the tube velocity is 5·5 ft. per second, and the air leaking into the vacuum system is 40 lb. per hour; the temperature of the condensate leaving the condenser is 96° F. Find the surface necessary in the air cooler to ensure that the air and vapour passing to the air ejector has a temperature of not more than 87° F. The vacuum to be maintained at the bottom of the condenser is 28·25 in. There are three hundred $\frac{3}{4}$-in. O/D tubes in the cooler.

At the inlet to the cooler

$$\begin{aligned} p_{1a} &= p_1 - p_{1s} \\ &= (30 - 28{\cdot}25) - 1{\cdot}7 \text{ (from steam tables)} \\ &= 1{\cdot}75 - 1{\cdot}7 \\ &= 0{\cdot}05 \text{ in. Hg.} \end{aligned}$$

At the outlet from the cooler we have (assuming there is no sensible loss in vacuum across the cooler)

$$\begin{aligned} p_{2a} &= p_2 - p_{2s} \\ &= 1{\cdot}75 - 1{\cdot}29 \\ &= 0{\cdot}46 \text{ in. Hg.} \end{aligned}$$

From equation (28) (p. 38) we have:

$$V_{1a} = [30 \times 12{\cdot}39(459{\cdot}6 + 96)]/[0{\cdot}05(459{\cdot}6 + 32)]$$
$$= (30 \times 12{\cdot}39 \times 555{\cdot}6)/(0{\cdot}05 \times 491{\cdot}6)$$
$$= 8400 \text{ c. ft. per pound of air,}$$

and $V_{1s} = 393{\cdot}4$ c. ft. per pound (from steam tables).

At outlet of cooler $V_{2a} = [30 \times 12{\cdot}39(459{\cdot}6 + 87)]/[0{\cdot}46(459{\cdot}6 + 32)]$

$$= (30 \times 12{\cdot}39 \times 546{\cdot}6)/(0{\cdot}46 \times 491{\cdot}6)$$
$$= 898 \text{ c. ft. per pound of air,}$$

and $V_{2s} = 513{\cdot}7$ c. ft. per pound (from steam tables).

Now the quantity of cooling water G flowing through the cooler is given by

$$G = (ANv \times 60 \times 6{\cdot}25)/144 \text{ gall. per minute,}$$

where A is area of each tube in square inches, N the number of tubes, and v the velocity through the tubes in feet per second,

i.e. $G = (0{\cdot}336 \times 300 \times 5{\cdot}5 \times 60 \times 6{\cdot}25)/144$

$$= 1444 \text{ gall. per minute.}$$

By equation (34) (p. 42), we find T_2.

$$600 \times 1444(T_2 - 80)$$
$$= 40[1038{\cdot}9(8400/393{\cdot}4 - 898/513{\cdot}7) + 0{\cdot}2375(96 - 87)];$$
$$866{,}400(T_2 - 80) = 40[1038{\cdot}9 \times 19{\cdot}6 + 2{\cdot}138];$$
$$T_2 = 0{\cdot}94 + 80$$
$$= 80{\cdot}94^\circ \text{ F.}$$

Then $\theta_m = [(96 - 80) - (87 - 80{\cdot}94)]/[2{\cdot}3 \times \log(16/6{\cdot}06)]$

$$= 9{\cdot}94/(2{\cdot}3 \times 0{\cdot}4216)$$
$$= 10{\cdot}25^\circ \text{ F.}$$

From equation (35) (p. 43) we get the necessary surface.

Now $R_w = C/v^{0{\cdot}82}$,

where C is the constant obtained from fig. 5 at the mean temperature of the cooling water, say, 80° F., and using $\frac{3}{4}$-in. O/D tubes,

i.e. $R_w = 2{\cdot}75/5{\cdot}5^{0{\cdot}82}$

$$= 0{\cdot}68.$$

Putting $R_a = 3$, and $R_t = 0{\cdot}08$, we get

$$S_c = [40(8400/393{\cdot}4 - 898/513{\cdot}7)(0{\cdot}68 + 0{\cdot}08 + 3)]/10{\cdot}25$$
$$= (40 \times 19{\cdot}6 \times 3{\cdot}76)/10{\cdot}25$$
$$= 289 \text{ sq. ft.}$$

Example.—During the trial trip of a merchant ship the following figures were taken from the main condenser:

Sea temperature	.. 50° F.
Condensate temperature	.. 70° F.
Vacuum at bottom of condenser ..	.. 29·1 in. Hg.
Vacuum at top of condenser ..	.. 28·8 in. Hg.

The air being dealt with by the air pump was measured to be 38 lb. per hour. The outlet temperature of the air and vapour leaving the air cooler was 55° F.

Find the total mass of air and vapour being dealt with by the air pump, and the ratio of the mass of air and vapour entering the cooler to the mass of air and vapour leaving the cooler; also find the mass of vapour condensed out by the cooler.

$$\begin{aligned}
\text{Now}\quad p_{1a} &= p_1 - p_{1s}\\
&= (30 - 29\cdot1) - 0\cdot739\\
&= 0\cdot161 \text{ in. Hg;}\\
p_{2a} &= (30 - 29\cdot1) - 0\cdot4375\\
&= 0\cdot4643 \text{ in. Hg;}\\
V_{1a} &= [30 \times 12\cdot39(459\cdot6 + 70)]/[0\cdot161(459\cdot6 + 32)]\\
&= 2487 \text{ c. ft. per pound of air;}\\
\text{and}\quad V_{1s} &= 871 \text{ c. ft. per pound;}\\
V_{2a} &= [30 \times 12\cdot39(459\cdot6 + 55)]/[0\cdot4643(459\cdot6 + 32)]\\
&= 840 \text{ c. ft. per pound of air;}\\
\text{and}\quad V_{2s} &= 1430 \text{ c. ft. per pound.}
\end{aligned}$$

From equation (29) (p. 38) we know that the total mass of air and vapour in any mixture is

$$M = m(1 + V_a/V_s),$$

therefore at the outlet from the cooler

$$\begin{aligned}
M_2 &= 38(1 + 840/1430)\\
&= 38 \times 1\cdot588\\
&= 60\cdot3 \text{ lb. per hour.}
\end{aligned}$$

But the air pump deals with the same mass as leaves the cooler, so that the air pump will deal with a mass of 60·3 lb. per hour of air and vapour at 55° F.

At the inlet to the cooler we have

$$\begin{aligned}
M_1 &= 38(1 + 2487/871)\\
&= 38 \times 3\cdot85\\
&= 146\cdot3 \text{ lb. per hour.}
\end{aligned}$$

The ratio will therefore be $146\cdot3/60\cdot3 = 2\cdot427$.

The mass of vapour condensed out by the cooler is obtained from equation (32) p. 42,

$$\begin{aligned} M_s &= m(V_{1a}/V_{1s} - V_{2a}/V_{2s}) \\ &= 38(2487/871 - 840/1430) \\ &= 38 \times 2{\cdot}269 \\ &= 86 \text{ lb. per hour.} \end{aligned}$$

Example.—During the test of a surface condenser it was desired to know the extent of the air leakage. For this purpose the following readings were taken:

Vacuum at bottom of condenser	28·8 in. Hg.
Barometer	29·9 in. Hg.
Temperature of condensate	80° F.
Outlet temperature of air and vapour from air cooler ..	70° F.
Inlet temperature of cooling water entering air cooler ..	60° F.
Outlet temperature of cooling water leaving air cooler ..	62° F.

The quantity of cooling water flowing through the cooler was estimated to be 1500 gall. per minute.

First find V_{1a} and V_{2a}, and from the steam tables find V_{1s} and V_{2s} and L.

Since the barometer is 29·9 in. Hg, the absolute pressure

$$\begin{aligned} p_1 &= 29{\cdot}9 - 28{\cdot}8 \\ &= 1{\cdot}1 \text{ in. Hg.} \\ \therefore\ p_{1a} &= 1{\cdot}1 - 1{\cdot}029 \\ &= 0{\cdot}071 \text{ in. Hg.} \\ \text{Then} \qquad V_{1a} &= [30 \times 12{\cdot}39(459{\cdot}6 + 80)]/[0{\cdot}071(459{\cdot}6 + 32)] \\ &= 5745 \text{ c. ft. per pound of air;} \\ \text{and } V_{1s} &= 636{\cdot}8 \text{ c. ft. per pound;} \\ p_{2a} &= p_2 - p_{2s} \\ &= 1{\cdot}1 - 0{\cdot}739 \\ &= 0{\cdot}361 \text{ in. Hg;} \\ V_{2a} &= [30 \times 12{\cdot}39(459{\cdot}6 + 70)]/[0{\cdot}361(459{\cdot}6 + 32)] \\ &= 1109 \text{ c. ft. per pound of air;} \\ \text{and } V_{2s} &= 871 \text{ c. ft. per pound.} \end{aligned}$$

The latent heat of the vapour at pressure p_{1s} is 1046·7. Then, substituting in equation (34) (p. 42), we get

$$\begin{aligned} &600 \times 1500(62 - 60) \\ &\qquad = m[1046{\cdot}7(5745/636{\cdot}8 - 1109/871) + 0{\cdot}2375(80 - 70)], \\ &\text{i.e. } m = 222 \text{ lb. per hour.} \end{aligned}$$

The mean temperature difference θ_m will be by equation (8):

$$\begin{aligned}\theta_m &= (\theta_1 - \theta_2)/[2{\cdot}3 \log_{10}(\theta_1/\theta_2)] \\ &= (16 - 8)/(2{\cdot}3 \log_{10}2) \\ &= 11{\cdot}57^\circ \text{ F.}\end{aligned}$$

It is best next to fix, provisionally, a tube velocity and find the condensation rate and surface, then work out the number of tubes to give the above velocity; thereafter finding the tube length and number of flows. If the condenser works out too long or too short, the tube velocity must be modified or the number of flows may be modified to give a reasonable length to the condenser. Sometimes the tube length is specified, and in such cases the tube velocity and surface must be modified to suit the tube length. The method is more or less trial and error, although judgment and experience may do a lot.

Let us therefore provisionally fix the tube velocity at 7 ft. per second.

Then the condensation rate w from equation (13) (p. 9) will be

$$w = \theta_m/[\text{R}_w + \text{R}_t + (\text{R}_f + \text{R}_a)],$$

and from the equation (14) (p. 9),

$$\begin{aligned}\text{R}_w &= \text{C}/v^{0{\cdot}82} \\ &= 3{\cdot}3/7^{0{\cdot}82} \\ &= 0{\cdot}67.\end{aligned}$$

Taking the value of $\text{R}_f + \text{R}_a = 1{\cdot}05$, and from p. 9, $\text{R}_t = 0{\cdot}08$ so that by substituting the values

$$\begin{aligned}w &= 11{\cdot}57/(0{\cdot}67 + 0{\cdot}08 + 1{\cdot}05) \\ &= 6{\cdot}43 \text{ lb. per square foot.}\end{aligned}$$

The total condensing surface necessary then will be

$$\begin{aligned}\text{S} &= 150{,}000/6{\cdot}43 \\ &= 23{,}330 \text{ sq. ft.}\end{aligned}$$

From p. 32 we have:

$$(n\text{A}v)/144 = \text{G}/(60 \times 6{\cdot}25),$$

where n is the number of tubes per flow,

$$\begin{aligned}\text{i.e. } n &= (144\text{G})/(\text{A}v \times 60 \times 6{\cdot}25) \\ &= (144 \times 19{,}230)/(0{\cdot}336 \times 7 \times 60 \times 6{\cdot}25) \\ &= 3139 \text{ tubes per flow.}\end{aligned}$$

Since the condenser is two flow, the total number of tubes will then be = 6278.

To find the length of the tube between the tube plates we have:

$$\begin{aligned} l &= (12\text{S})/(2\pi \text{D}n) \text{ (see p. 32)} \\ &= (12 \times 23{,}330 \times 4)/(2 \times 3139 \times \pi \times 3) \\ &= 18\cdot 9 \text{ ft.} \end{aligned}$$

If this length is excessive for the convenient withdrawal of the tubes then the tube velocity must be reduced. This would reduce the length l, although the surface S would be increased. The length of 18·9 ft. is, however, not abnormal.

To find the friction head across the condenser we must know the overall length of the tubes. Assuming the tube plates $1\frac{1}{4}$ in. thick, and the tubes to project $\frac{1}{8}$ in. beyond the tube plates, the overall length of the tubes will be 18 ft. 10·8 in. + 2·5 in. + 0·25 in. = 19 ft. 1·55 in. = say, 19 ft. $1\frac{1}{2}$ in.

From fig. 7 we find that the friction head in a 10-ft. length, at a velocity of 7 ft. per second, is 4·35 ft., and the entrance friction head is 1·2 ft.

Since the condenser is two flow, the value of the condenser friction H_c (see p. 19) will be 1 ft. 6 in.

The total friction head H will be (see p. 17)

$$\begin{aligned} \text{H} &= 2\text{H}_t + 2\text{H}_e + \text{H}_c \\ &= (2 \times 4\cdot 35 \times 19\cdot 125)/10 + (2 \times 1\cdot 2) + 1\cdot 6 \\ &= 16\cdot 6 + 2\cdot 4 + 1\cdot 6 \\ &= 20\cdot 6 \text{ ft.} \end{aligned}$$

This is under the allowance of 22 ft.

To fix up the surface necessary in the air cooler it is usually easiest to fix arbitrarily the surface by taking it as a percentage of the condensing surface, and finding the quantity of air that this surface is capable of cooling to an arbitrarily fixed air outlet temperature. The correct surface can usually be found at the second trial.

Let us take the cooler surface S_c as 5 per cent of the condensing surface (a common value), and let us fix the outlet temperature of the air and vapour at 5° F. higher than the cooling water inlet temperature, i.e. 55 + 5 = 60° F.

The surface then will be

$$\begin{aligned} \text{S}_c &= 23{,}330/20 \\ &= 1166\cdot 5 \text{ sq. ft.} \end{aligned}$$

The air and vapour velocity through the cooler is usually very low, so that the pressure drop becomes negligible; the absolute pressure p throughout the cooler therefore will be

$$30 - 29\cdot 21 = 0\cdot 79 \text{ in Hg.}$$

Then at inlet:

$$\begin{aligned} & p_{1a} = p - p_{1s}, \\ \text{i.e.} \quad & p_{1a} = 0\cdot 79 - 0\cdot 764 \\ & \phantom{p_{1a}} = 0\cdot 026 \text{ in. Hg;} \end{aligned}$$

and at outlet:

$$
\begin{aligned}
p_{2a} &= p - p_{2s} \\
&= 0{\cdot}79 - 0{\cdot}522 \\
&= 0{\cdot}268 \text{ in. Hg;} \\
\text{also}\quad V_{1a} &= [30 \times 12{\cdot}39(459{\cdot}6 + 71)]/[0{\cdot}026(459{\cdot}6 + 32)] \\
&= 15{,}430 \text{ c. ft. per pound;} \\
\text{and}\quad V_{1s} &= 843 \text{ c. ft. per pound;} \\
V_{2a} &= [30 \times 12{\cdot}39(459{\cdot}6 + 60)]/[0{\cdot}268(459{\cdot}6 + 32)] \\
&= 1466 \text{ c. ft. per pound;} \\
\text{and}\quad V_{2s} &= 1208 \text{ c. ft. per pound.}
\end{aligned}
$$

It simplifies the calculation here to take the arithmetic mean temperature difference in equation (35) instead of the logarithmic mean temperature difference without appreciable error, so that equation (35) becomes

$$S_c = [m(V_{1a}/V_{1s} - V_{2a}/V_{2s})(R_w + R_t + R_a)]/[\{(t_1 - T_1) + (t_2 - T_2)\}/2],$$

$R_w = 0{\cdot}67$ (see p. 49).

Put $R_t = 0{\cdot}08$; for air cooler conditions R_a may be taken as $4{\cdot}5$.

$$
\begin{aligned}
1166{\cdot}5 &= [2m(15{,}430/843 - 1466/1208)(0{\cdot}67 + 0{\cdot}08 + 4{\cdot}5)] \\
&\qquad \div [(71 - 55) + (60 - T_2)] \\
&= (2m \times 17{\cdot}086 \times 5{\cdot}25)/(76 - T_2); \\
\text{or}\quad T_2 &= (179{\cdot}403m - 88664)/1166{\cdot}5 \\
&= -0{\cdot}154m + 76.
\end{aligned}
$$

Since the tube velocity in the air cooler is the same as that in the condenser, the quantity of cooling water flowing through the cooler will be proportional to half the condensing surface (the condenser being two flow) and the whole of the cooler surface,

$$
\begin{aligned}
\text{i.e.}\quad G &= (19{,}230 \times 1166{\cdot}5 \times 2)/23{,}330 \\
&= 1923 \text{ gall. per minute.}
\end{aligned}
$$

Again, from equation (34) (p. 42) we have, by substitution:

$$
\begin{aligned}
&600 \times 1923(T_2 - 55) \\
&\qquad = m[1051{\cdot}7\{15430/843 - 14466/1208\} + 0{\cdot}2375(71 - 60)], \\
\text{or}\quad T_2 &= [m(1051{\cdot}7 \times 17{\cdot}086 + 2{\cdot}613)/(600 \times 1923)] + 55 \\
&= 0{\cdot}016m + 55.
\end{aligned}
$$

Solving these two equations connecting T_2 and m, we get:

$$
\begin{aligned}
-0{\cdot}154m + 76 &= 0{\cdot}016m + 55, \\
\therefore\ m &= 123{\cdot}5 \text{ lb. of air per hour.}
\end{aligned}
$$

Now the allowable air leakage would be obtained by the B.E.A.M.A rule (see p. 37).

$$\begin{aligned} m &= (0{\cdot}5W)/1000 + 3 \\ &= (0{\cdot}5 \times 150{,}000)/1000 + 3 \\ &= 75 + 3 \\ &= 78 \text{ lb. of air per hour.} \end{aligned}$$

An air cooler surface, therefore, of 5 per cent of the condensing surface is ample to deal with the allowable air leakage.

Since the tubes in the air cooler must, for constructional reasons, be the same length as the tubes in the condenser the number in the cooler will be

$$\begin{aligned} &= \frac{\text{number of tubes in condensing portion} \times \text{surface of air cooler}}{\text{total condensing surface}} \\ &= (2 \times 3139 \times 1166{\cdot}5)/23{,}330 \\ &= \text{say, } 314. \end{aligned}$$

It is usual to arrange for the same tube velocity in both passes, which means that the number of tubes in the first pass of the condensing portion, plus the number of tubes in the cooler, must equal the number of tubes in the second pass.

Arranging the tubes then in this fashion we get the number in each pass to be

$$\begin{aligned} &= (2 \times 3139 + 314)/2 \\ &= 3296. \end{aligned}$$

The actual tube velocity will then be

$$\begin{aligned} &= [(19{,}230 + 1923) \times 10 \times 144]/[60 \times 62{\cdot}5 \times 0{\cdot}336 \times 3296] \\ &= 7{\cdot}33 \text{ ft. per second.} \end{aligned}$$

The velocity we provisionally fixed upon was 7 ft. per second. This increased velocity will give a slightly increased margin to the condensing surface, and will slightly increase the friction head across the condenser, which now becomes 22·4 ft., or just slightly over the specified head.

The principal data for this condenser may now be collected as follows:

Duty, 150,000 lb. steam per hour.
Vacuum at steam inlet branch, 29·1 in. Hg.
Barometer, 30 in. Hg.
Inlet temperature of cooling water, 55° F.
Quantity of cooling water required, 19,230 + 1923 = 21,153 gall. per minute.
Surface in condenser, 23,330 sq. ft.
Surface in cooler, 1166·5 sq. ft.
Friction head across condenser, 22·4 ft.
Pressure drop in steam space, 0·11 in. Hg.

Tube velocity, 7·33 ft. per second.
Outlet temperature of air and vapour from cooler, 60° F.
Length between tube plates, 18 ft. 9 in.
Number of tubes in condenser, 6278.
Number of tubes in cooler, 314.
Tubes, $\frac{3}{4}$ in. O/D 18, S.W.G. thick.

From these particulars the condenser may be set down on the drawing-board, when it only remains to so space the tubes at the top that the entering steam speed does not exceed that corresponding to the pressure drop of 0·11 in. Hg. This steam speed is obtained from fig. 12, 198 ft. per second.

We shall now take an example of a steam condenser for the reciprocating engine of a cargo boat of 3000 i.h.p.

Let the given conditions be:

Sea temperature, 60° F.
Vacuum, 27 in. with a 30-in. Hg barometer.
Tubes to be $\frac{3}{4}$ in. O/D, 18 S.W.G. thick, and 9 ft. overall length.
Steam consumption of engine, say, 15 lb. per i.h.p. hour.

For reasons which will be given later, no air cooler need be fitted to a condenser for 27 in. vacuum. We shall, therefore, fit no air cooler to this condenser. The steam entering the condenser will be

$$= 15 \times 3000$$
$$= 45{,}000 \text{ lb. per hour.}$$

Let fig. 20 represent our condenser. Since the tube length is specified it will be necessary to rough out the particulars provisionally from assumed values.

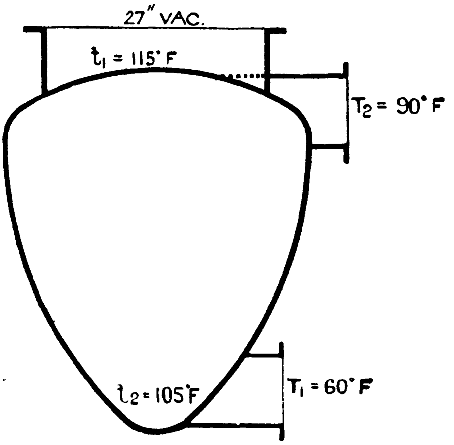


Fig. 20

Let these assumed values be:

Condenser to be three flow.
Tube velocity, 6 ft. per second.
Temperature difference between ingoing steam and outgoing cooling water (see p. 15), 25° F.
The temperature drop across the steam space, $(t_1 - t_2) = 10°$ F.

The outlet temperature of the cooling water will be

$$T_2 = 115 - 25$$
$$= 90° \text{ F.},$$
$$\text{and } t_2 = 115 - 10$$
$$= 105° \text{ F.}$$

Then the mean temperature difference θ_m will be

$$\begin{aligned}\theta_m &= (\theta_1 - \theta_2)/[2{\cdot}3 \log_{10}(\theta_1/\theta_2)] \\ &= [(105 - 60) - (115 - 90)]/[2{\cdot}3 \log_{10}(45/25)] \\ &= 20/(2{\cdot}3 \times 0{\cdot}255) \\ &= 34^\circ \text{ F}.\end{aligned}$$

The quantity of cooling water is given by equation (16) (p. 16) and taking $q = 0{\cdot}9$,

$$\begin{aligned}\text{G} &= [45{,}000(0{\cdot}9 \times 1027 + 115 - 105)]/[600(90 - 60)] \\ &= (45{,}000 \times 934)/(600 \times 30) \\ &= 2335 \text{ gall. per minute.} \\ \text{Then } w &= \theta_m/(\text{R}_w + \text{R}_t + \text{R}_f + \text{R}_a), \\ \text{and } \text{R}_w &= \text{C}/v^{0{\cdot}82} \\ &= 2{\cdot}72/6^{0{\cdot}82} \\ &= 0{\cdot}626.\end{aligned}$$

Putting $\text{R}_f + \text{R}_a = 1{\cdot}3$ (p. 10), we get

$$\begin{aligned}w &= 34/(0{\cdot}626 + 0{\cdot}08 + 1{\cdot}3) \\ &= 17 \text{ lb. per square foot.}\end{aligned}$$

The provisional condensing surface therefore is

$$\begin{aligned}&= 45{,}000/17 \\ &= 2650 \text{ sq. ft.}\end{aligned}$$

Assuming tube plates 1 in. thick, and the tubes projecting $\frac{1}{8}$ in. beyond the tube plates, the length l between the tube plates will be

$$\begin{aligned}l &= 9 \text{ ft.} - 2\tfrac{1}{4} \text{ in.} \\ &= 8 \text{ ft. } 9\tfrac{3}{4} \text{ in.}\end{aligned}$$

The number of tubes per flow will be

$$\begin{aligned}n &= (12\text{S})/(3\pi \text{D}l) \text{ (see p. 32)} \\ &= (12 \times 2650 \times 4)/(3 \times \pi \times 3 \times 8{\cdot}812) \\ &= 510.\end{aligned}$$

Since there are three passes the total number of tubes will be

$$\begin{aligned}&= 510 \times 3 \\ &= 1530.\end{aligned}$$

The actual tube velocity is

$$\begin{aligned}v &= (2335 \times 144)/(6{\cdot}25 \times 0{\cdot}336 \times 510 \times 60) \\ &= 5{\cdot}23 \text{ ft. per second.}\end{aligned}$$

The provisionally fixed tube velocity was 6 ft. per second, so that either the surface must be slightly increased or the quantity of cooling water passing must be slightly decreased.

Let the surface be increased to 2800 sq. ft., then the actual condensation rate will be

$$\begin{aligned} w &= 45{,}000/2800 \\ &= 16{\cdot}07 \text{ lb. per square foot.} \end{aligned}$$

The number of tubes per flow will be increased to

$$\begin{aligned} &= (510 \times 2800)/2650 \\ &= 539, \end{aligned}$$

and the tube velocity will be decreased to

$$\begin{aligned} &= (5{\cdot}23 \times 510)/539 \\ &= 4{\cdot}95 \text{ ft. per second.} \end{aligned}$$

Then the condensation rate which the condenser is capable of giving is:

$$\begin{aligned} w &= 34/(R_w + 0{\cdot}08 + 1{\cdot}3), \\ \text{and } R_w &= 2{\cdot}72/4{\cdot}95^{0{\cdot}82} \\ &= 2{\cdot}72/3{\cdot}71 \\ &= 0{\cdot}733. \\ \therefore\ w &= 34/(0{\cdot}733 + 0{\cdot}08 + 1{\cdot}3) \\ &= 34/2{\cdot}113 \\ &= 16{\cdot}09 \text{ lb. per square inch,} \end{aligned}$$

which is almost the same as the actual rate of 16·07.

The friction head across the condenser will be

$$\begin{aligned} H &= 3H_t + 3H_e + H_c \\ &= (3 \times 2{\cdot}1 \times 9)/10 + 3 \times 0{\cdot}65 + 2 \\ &= 5{\cdot}67 + 1{\cdot}95 + 2 \\ &= 9{\cdot}62 \text{ ft.} \end{aligned}$$

The principal data for this condenser may now be collected, and is as follows:

Duty, 45,000 lb. steam per hour.
Vacuum at steam inlet branch, 27 in. Hg.
Barometer, 30 in. Hg.
Inlet temperature of cooling water, 60° F.
Quantity of cooling water required, 2315 gall. per minute.
Condensing surface, 2800 sq. ft.
Friction head across condenser, 9·62 ft.
Tube velocity, 4·95 ft. per second.
Length between tube plates, 8 ft. 9¾ in.
Number of tubes in condenser, 1617.
Tubes, ¾ in. O/D, 18 S.W.G. thick.

CONSIDERATIONS WHICH SHOW THAT ONLY HIGH VACUUM CONDENSERS NEED BE FITTED WITH AN AIR COOLER

Let the conditions at the bottom of the low vacuum condenser be:

Vacuum, 25·5 in. with a 30-in. Hg barometer,
Condensate temperature, 126° F.,

and let the conditions at the bottom of the high vacuum condenser be:

Vacuum, 28·5 in. with a 30-in. Hg barometer,
Condensate temperature, 79° F.

In the low vacuum condenser the partial air pressure will be

$$\begin{aligned} p_{1a} &= p_1 - p_{1s} \\ &= (30 - 25\cdot5 \text{ in.}) - 4 \text{ in.} \\ &= 0\cdot5 \text{ in. Hg,} \end{aligned}$$

and in the high vacuum condenser the partial air pressure will be

$$\begin{aligned} p_{2a} &= p_2 - p_{2s} \\ &= (30 - 28\cdot5 \text{ in.}) - 1 \text{ in.} \\ &= 0\cdot5 \text{ in. Hg.} \end{aligned}$$

There is thus a loss in vacuum due to air in each condenser of 0·5 in. Hg.

The corresponding loss in condensate temperature is, for the low vacuum condenser,

= vacuum temperature at 25·5 in. Hg — vacuum temperature at 26 in. Hg
= 130 — 126
= 4° F.

The corresponding loss for the high vacuum condenser is

= vacuum temperature at 28·5 in. Hg — vacuum temperature at 29 in. Hg
= 92 — 79
= 13° F.

Therefore, if no air cooler be fitted to either condenser, the condensate must be cooled 4° F. below the vacuum temperature in the low vacuum condenser, and 13° F. in the high vacuum condenser to give an equal partial air pressure in each case of 0·5 in.

This depression of the condensate temperature is a direct heat loss, as the loss has to be made up in the boiler fuel and represents an increased fuel consumption in the furnaces of approximately

$(4 \times 100)/1000 = 0\cdot4$ per cent in the low vacuum installation,
and $(13 \times 100)/1000 = 1\cdot3$ per cent in the high vacuum installation.

The latter installation, therefore, has the greater heat loss, and it is to minimize this loss that an air cooler is fitted.

The above comparison is made on equal air pressure conditions in the condenser and is thus a true comparison of the condensers themselves. There is, however, the secondary consideration of the air-pumping capacity of the air pumps fitted in the two cases, which if not equal would affect the air pressures. These pressures can be so arranged that the heat loss would be the same, but in this case more energy must be expended by the air pump to maintain the lower air pressure in the high vacuum condenser. We shall discuss this point further when comparing displacement air pumps with air ejectors.

AREA OF EXHAUST INLET

The allowable speed of the steam through the inlet should vary with the vacuum, and the length of exhaust or eduction pipe. From considerations similar to these investigated on pp. 25 to 28 for the pressure drop across a surface condenser, this allowable steam speed should increase with increase in vacuum. Owing to increase of friction with increased length of piping the allowable steam speed should decrease with increased length of piping. For each installation, therefore, the steam speed allowed should take into consideration the length and number of bends in the exhaust piping as well as the vacuum. With the modern underhung condenser for turbines the length of the eduction pipe is reduced to a minimum, so that the vacuum loss from this cause is reduced to a minimum. With reciprocating engines there is usually a large amount of throttling in the exhaust ports and passages, but with these engines the drop in efficiency from loss of vacuum caused by throttling is much lower than it is for turbines.

For normal lengths of exhaust piping the area of the steam inlet branch on the condenser may be found from the following rule, which gives average working values.

$A = W/X,$

where A is the area of steam inlet in square inches;

W, the weight of steam in the condenser in pounds per hour; and

X is a constant taken from Table II.

TABLE II

Vacuum, inches Hg.	24	24·5	25	25·5	26	26·5	27	27·5	28	28·5	29
For Land Reciprocating Engines, X ..	102	95	89	83	76	69	61	53	44	—	—
For Turbines, X ..	—	—	—	—	—	—	20	17·5	15	12·5	10
For Marine Reciprocating Engines, X	170	158	148	138	126	115	102	90	73	—	—

MISCELLANEOUS EXAMPLES ON SURFACE CONDENSERS

1. A marine surface condenser was reported to be giving an unsatisfactory vacuum, and to locate the fault the ship's engineers were requested to make the following observation at the designed H.P. nozzle box steam pressure, and report. The observations were:

Vacuum	26 in. Hg.
Barometer	29·75 in. Hg.
Sea temperature	55° F.
Outlet temperature of cooling water ..	73° F.
Temperature of condensate	100° F.

The designed conditions were:

Vacuum	28 in. Hg.
Barometer	30 in. Hg.
Sea temperature	60° F.
Outlet temperature of cooling water ..	80° F.
Temperature of condensate	95° F.

State whether the fault was air leakage or a dirty condenser, or a deficiency in cooling water.

Answer.—The rise in the temperature of the cooling water is 18° F. as against the designed rise of 20° F.; that is less than the designed rise. Since the designed quantity of steam is being condensed, because the H.P. nozzle box steam pressure is the designed pressure, there is thus more cooling water passing than the designed quantity and therefore no deficiency in cooling water. This assures equal condensate temperatures, but since the difference is only 5° F. of sensible heat it may be here neglected.

The difference between the condensate temperature and the sea temperature is 45° F. against the designed difference of 35° F. Bearing in mind the fact that there is 18° F. rise in the cooling as against the designed rise of 20° F., the above increase of difference from 35° F. to 45° F. indicates an increase in the resistance to heat flow, due probably to a slight deposit of scale inside the tubes, but not sufficient to account for all the vacuum loss. The vacuum corresponding to a condensate temperature of 100° F. is 28·07 in. Hg, while the vacuum recorded corrected to a 30-in. Hg barometer is 26·52 in. Hg, so that the vacuum loss caused by air leakage is 28·07 — 26·25 = 1·82 in. Hg. The main fault therefore lies in abnormal air leakage. This difference between the actual vacuum in the condenser and the vacuum corresponding to the condensate temperature is an accurate air leakage gauge, and always indicates whether loss of vacuum is due to air leakage or not. If the vacuum recorded be that at the top of the condenser the vacuum loss from top to bottom should be taken into account, and to eliminate this loss it is always advisable in an investigation of this sort to fit a vacuum gauge, preferably of the sentinel type, to the bottom of the condenser.

2. A surface condenser, having 26,000 sq. ft. of condensing surface, is designed to record 28 in. vacuum at the exhaust inlet when supplied with 26,000 gall. of cooling water per minute, at a temperature of 80° F. The condenser is fitted to an 18,000-kw. turbo-alternator, having a steam consumption rate of 11 lb. per kilowatt-hour. The tubes are $\frac{3}{4}$ in. O/D, and the tube velocity is 6·5 ft. per second. The temperature of the condensate is 93° F. Calculate the value of the air resistance factor R_a.

Answer.—0·784.

3. In the above example 2 the steam velocity past the first row of tubes is 173 ft. per second when the vacuum is 28 in. Hg. Find the pressure drop from top to bottom of the steam space, and hence find the partial air pressure at the bottom of the condenser.

Answer.—Pressure drop = 0·2 in. Hg.
Partial air pressure = 0·22 in. Hg.

4. A surface condenser having 16,135 sq. ft. of condensing surface is designed to condense 120,000 lb. per hour of exhaust steam at 29 in. Hg vacuum. The cooling water has an inlet temperature of 50° F., and the designed rise in temperature of the cooling water is 16·67° F. Taking the condensate as being 10° F. lower than the vacuum temperature, calculate what the vacuum would be at inlet cooling water temperatures of 40° F., 60° F., 70° F., and 80° F., and plot a curve showing the relation between the vacuum and inlet temperature of the cooling water. Assume a constant steam flow, a constant quantity of cooling water and equal pressure drops from top to bottom in each case. The tubes are 1 in. outside diameter and the tube velocity is 6 ft. per second.

Answer.

Inlet temperature of cooling water ..	40° F.	60° F.	70° F.	80° F.
Vacuum	29·24 in.	28·65 in.	28·19 in.	27·63 in.

Note.—Owing to the variation in vacuum drop or pressure drop from the top to the bottom with actual vacuum, the vacuum recorded at 40° F. would be slightly less, and the vacua at 60°, 70°, and 80° F. slightly greater than the above, which have been calculated on the given assumption of equal vacuum drops.

5. Take the condenser given in examples 2 and 3, and estimate what the vacua would be at inlet cooling water temperatures of 90° F. and 70° F. and 60° F., taking account of the variation in pressure drop with vacuum.

Answer.

Inlet cooling water temperature	90° F.	70° F.	60° F.
Vacuum	27·34 in.	28·49 in.	28·9 in.

6. Find the difference in temperature between the top and bottom of the steam space in a surface condenser given the following:

Vacuum at top, 29 in. Hg.
Steam speed past first row of tubes, 240 ft. per second.
Partial air pressure at bottom, 0·15 in. Hg.

Answer.—10·7° F.

7. The condenser given in example 6 is supplied with hotter cooling water, so that the vacuum is reduced to 28 in. Hg while the steam flow is held constant. Find what the temperature difference between the top and the bottom would be under the new conditions.

Answer.—4·3° F.

8. Calculate the quantity of cooling water necessary to condense 100,000 lb. of exhaust steam per hour, having a wetness fraction of 0·08. The design of the condenser is such that the temperature difference between the top and the bottom is 8° F. The vacuum at the exhaust inlet is 28·75 in. Hg, and the inlet temperature of the cooling water is 65° F., and the outlet temperature 77° F.

Answer.—13,450 gall. per minute.

9. Calculate the condensing surface necessary to condense 35,000 lb. of exhaust steam per hour from a marine turbine and maintain a vacuum at the exhaust inlet of 28·5 in. Hg with sea water at 65° F. The temperature difference between the outgoing cooling water and the ingoing steam is 15° F., and the allowable pressure drop from top to bottom of the condenser must not exceed 0·25 in. Hg. The air pump is of such a capacity that it can reduce the partial air pressure to 0·3 in. Hg. The tubes are $\frac{3}{4}$ in. external diameter, and the tube velocity is 7 ft. per second. Take $R_a = 0{\cdot}95$.

Answer.—4121 sq. ft.

10. Calculate the condensing surface necessary to condense 18,000 lb. of exhaust steam from a reciprocating marine engine and maintain a vacuum of 26 in. Hg at the exhaust inlet with a sea temperature of 70° F. The heat given up by the steam per pound in condensing and cooling to the hotwell temperature is 965 B.Th.U. The ratio of the quantity of cooling water to the steam condensed is 30, and the pressure drop is 0·1 in. Hg. The partial air pressure at the bottom of the condenser is 0·45 in. Hg. The tubes are $\frac{3}{4}$ in. external diameter, and the tube velocity is 6 ft. per second. Take $R_a = 1{\cdot}4$.

Answer.—1187 sq. ft.

11. In example 9 find the mass of air and vapour being dealt with by the air pump per pound of air withdrawn. No air cooler is fitted to the condenser.

Answer.—2·965 lb. per pound of air.

12. Find the air cooler surface which should be provided in a surface condenser designed to condense 40,000 lb. of exhaust steam at 28·5 in. Hg vacuum. The pressure drop across the steam space is 0·2 in. Hg, and the partial air pressure at the bottom of the condenser is 0·05 in. Hg. The inlet temperature of the cooling water is 65° F., and the temperature rise of the cooling water passing through the cooler is 1° F. The partial air pressure at the air outlet from the cooler must not be less than 0·3 in. Hg. The tube velocity is 7 ft. per second, and tubes are ¾ in. O/D. Take $R_a = 3{\cdot}5$.

Answer.—64·19 sq. ft.

13. A surface condenser has 5210 ¾-in. external diameter tubes, 18 S.W.G. thick, and 18 ft. 6 in. long. The circulating water makes two passes through the condenser, and the quantity passing is 16,000 gall. per minute. Estimate what the friction head would be.

If the difference in level between the inlet and outlet branches be 7 ft., what would be the difference in recorded pressure between two pressure gauges, one fitted on the centre line of each branch.

Answer.—19·82 ft. friction head. 25·72 ft. or 11·2 lb. per square inch difference.

Various Makes of Surface Condensers

WEIR UNIFLUX SURFACE CONDENSER

Fig. 21 illustrates the latest design of the "Weir Uniflux" high vacuum condenser for land power stations.

The condenser is of the two-flow type. The cooling water enters at the inlet A and flows through the bottom nest of tubes to the return end B, where the direction of flow is reversed. The water returns through the top nest of tubes and leaves by the outlet C. The steam enters at D and flowing vertically downwards across the cold tubes, becomes condensed; the condensate is extracted through the branch E. The air, vapour, and non-condensible gases pass to the integral air cooler and devaporizer F, and entering the cooler at the bottom flow upwards to the air outlet G; a large percentage of the vapour being condensed, and the air and non-condensible gases considerably cooled.

At the steam inlet the tubes are pitched wide apart, and the pitching is reduced in steps from the top to the bottom of the condenser, thereby giving an approximately uniform speed to the steam as it flows from the top to the bottom. The wide shallow shape of this condenser will be noticed, and the consequent short penetration in the tube bank, coupled with the diminution of the tube pitching, render the pressure drop from the top to the bottom of the steam space very small, and the depression of the condensate temperature below the vacuum temperature within a few degrees.

The steam inlet is made the full breadth of the condenser, and as long as the construction will permit, and, if the turbine exhaust branch cannot conveniently be made this size, an adaptor piece is fitted between the turbine exhaust branch and the condenser. By this means deflector baffle plates are dispensed with and the whole tube surface steam swept, so that no stagnant zones exist where air may collect and render the tube surface inefficient. For this reason the condensation rate in this condenser is very high. The integral air cooler F extends from end to end of the steam space, and is formed by a baffle extending between the tube-supporting diaphragm plates; angle iron flanges are riveted to the baffle and bolted to the diaphragm, tube plates, and shell plates. A proper joint is made with jointing material, so that steam cannot leak from the steam space into the top of the cooler space and thereby render the cooler surface ineffective. With this construction the air and vapour must pass round the bottom of the baffle, so that the whole cooler surface is effective. The dry air outlet G is placed a little lower than the top of the cooler, and three or four rows of tubes are arranged between the underside of the baffle and the top of the air branch. This ensures that the air, which has already been cooled in passing through the cooler, is not reheated when passing into the air suction branch by flowing over the baffle, the baffle being maintained relatively hot by its being in contact with the steam on its steam side. The air is thus effectively cooled, and once cooled is protected against reheating by its coming into contact with a relatively hot baffle just as it enters the air suction branch. A hole is cut through each diaphragm plate as shown at H, so that each compartment of the cooler is effective.

When the conditions of space, owing to the arrangement of the turbine foundations or seatings, permit, the makers sometime fit an independent air-suction branch to each compartment, and connect these branches externally by a branch pipe terminating in a common branch, which is led to the air ejector suction. Either arrangement is quite effective, the former having the advantage of simplicity and lower manufacturing costs.

The tube-supporting diaphragms are made of mild steel, and are spaced so that the unsupported length of tube is normally not more than 4 ft., although in special cases condensers have been successfully installed with an unsupported tube length of 5 ft. The question of the unsupported tube length is one of vibration, and if the natural period of the unsupported length of tube happens to synchronize with the vibration of the turbine the tubes vibrate badly and ultimately split. This trouble is aggravated if the clearance between the tube hole in the diaphragm plates and the tubes is excessive, and present-day practice is to have this clearance as small as possible, so that the tube is well held at each diaphragm plate. A normal clearance is $\frac{20}{1000}$ in. The device has been tried of placing the diaphragms out of line with the tube plates so that when the tubes are in place they are hogged; commercially straight condenser tubes, however, are usually off the truth to a sufficient extent to give the above effect with diaphragm plates and tube plates in line. The diaphragm plates are fitted clear of the condenser shell by about $\frac{1}{4}$ in., the attachment being by mild steel lugs riveted

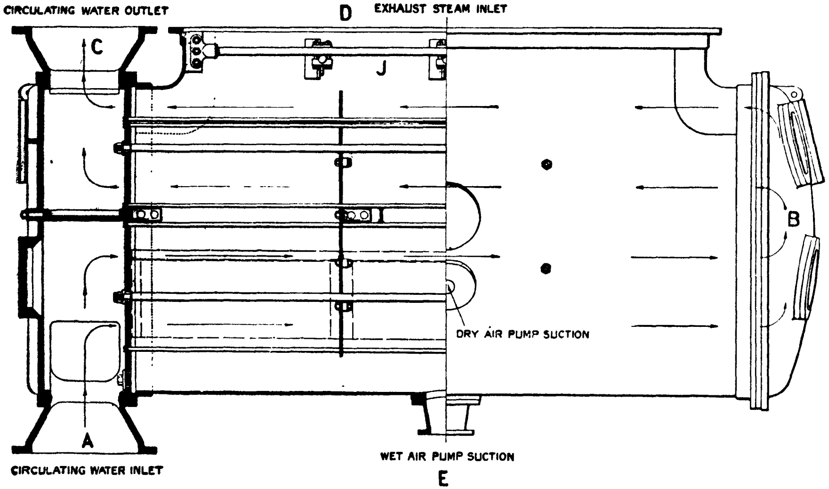
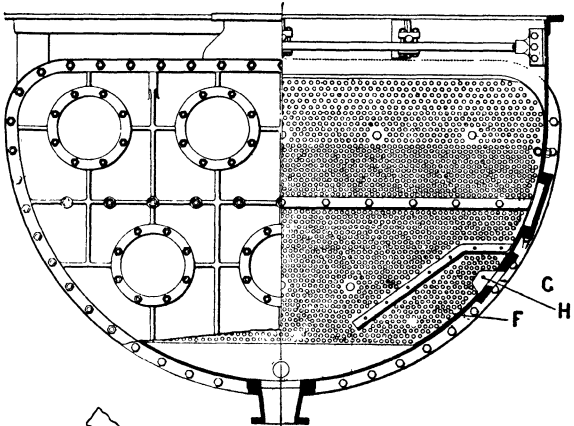


Fig. 21



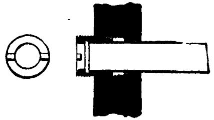
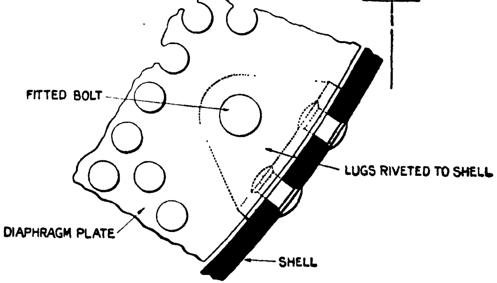


Fig. 23



Weir Uniflux" Condenser

Fig. 22

to the shell and a fitted bolt passing through the diaphragm plate and the lug; one of these lugs is shown in fig. 22.

The reason for this construction is that the diaphragm is used as a support to stay the sides of the shell against the collapsing pressure due to the vacuum; the diaphragm is a strut and the fitted bolts passing through the lugs transfer some of the load on to the diaphragm. The diaphragm is stiffened laterally by the diaphragm stiffeners I (fig. 21). The exhaust steam inlet branch is stayed and stiffened by the palm stays J, the cross stays being clamped to the longitudinal stays by glands, thus reducing their virtual length as struts. The tube plates are stayed by long round steel stays which pass between them, the ends being reduced where they pass through the tube plates so as to form a shoulder against which the tube plates are tightened. The ends of the steel stays are protected and leakage of circulating water guarded against by the brass cap nuts over the ends of the stays. A steel angle stiffener is fitted behind the tube plates, this angle is studded, the studs passing through the tube plate at the return water end, and through both the tube plate and the division plate at the circulating water inlet end. The shell of the condenser is of mild steel plate riveted and caulked, while the end flanges and the exhaust inlet flange are steel castings. The water boxes and covers are of cast iron. The return end cover is dished, while the flat cover on the water box is suitably ribbed to withstand the water pressure. Large cast-iron inspection doors are fitted to these end covers, so that all the tube ends can be inspected without removing the end covers The tube plates are of rolled yellow metal, having a composition of 60 per

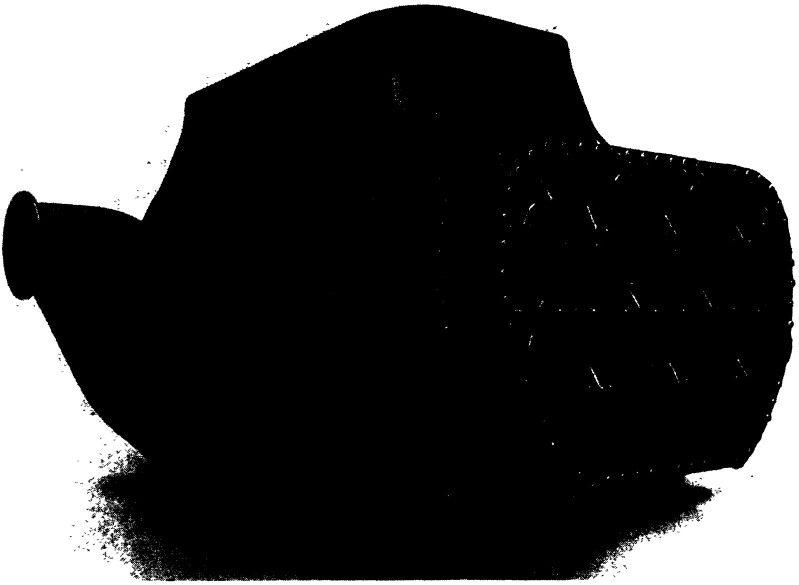


Fig. 24.—" Weir Uniflux " Condensers

cent copper, 40 per cent zinc; while the tubes are of brass, having a composition of 70 per cent copper, 30 per cent zinc, and are ¾ in. outside diameter, 18 S.W.G. thick.

To allow for the differential expansion between the shell and tubes, the tubes are ferruled into the tube plates at both ends and packed with linen tape packings. Fig. 23 shows a section through one of the tube ends and ferrules. The ferrules have an inside lip to prevent the tubes creeping axially and ultimately coming out. The inlet side of the ferrule is rounded off to reduce the vena contracta at entrance and thereby reduce the friction loss at entrance.

Fig. 24 shows one of three " Weir Uniflux " condensers recently installed at the Birmingham Corporation generating station at Nechells. These condensers were designed to maintain a vacuum of 28 in. Hg, with a cooling water inlet temperature of 80° F., when dealing with 165,000 lb. of exhaust steam per hour from a 15,000-kw., Fraser & Chalmers turbo-alternator.

TABLE III

TEST FIGURES OF 15,000-KW. CONDENSING PLANT AT NECHELLS GENERATING STATION, BIRMINGHAM

Load in Kilowatts.	Steam Load in Pounds per Hour.	Barometer in Inches of Hg.	Vacuum at top of Condenser.	Vacuum at Ejector Suction.	Temperatures in Degrees Fahrenheit		
					Inlet Cooling Water.	Outlet Cooling Water.	Condensate.
18,360	195,600	30	28·02	28·12	75	93	93
15,000	164,000	30	28·11	28·19	78	93	92
13,300	149,000	30	28·21	28·29	76	91	91
7,500	89,000	30	28·43	28·46	77	85	86
3,750	51,500	30	28·5	28·6	75	84	83

The test sheet figures of this condensing plant are given in Table III, and the author is indebted to Messrs. G. & J. Weir, Ltd., for permission to publish these.

MIRRLEES WATSON SURFACE CONDENSER

Figs. 25, 26, and 27 show the latest design of Messrs. The Mirrlees Watson Company, for a large surface condenser suitable for a 15,000-kw. turbo-alternator. The internal tube arrangement of this condenser is very similar to that described for the " Weir Uniflux " condenser, excepting that the top nest of tubes is arranged for square pitching instead of diagonal pitching. The pitching at the top is very wide and gradually decreases from top to bottom of the condenser. A devaporizing chamber is also fitted.

The designed conditions of this condenser are: steam duty, 160,000 lb.

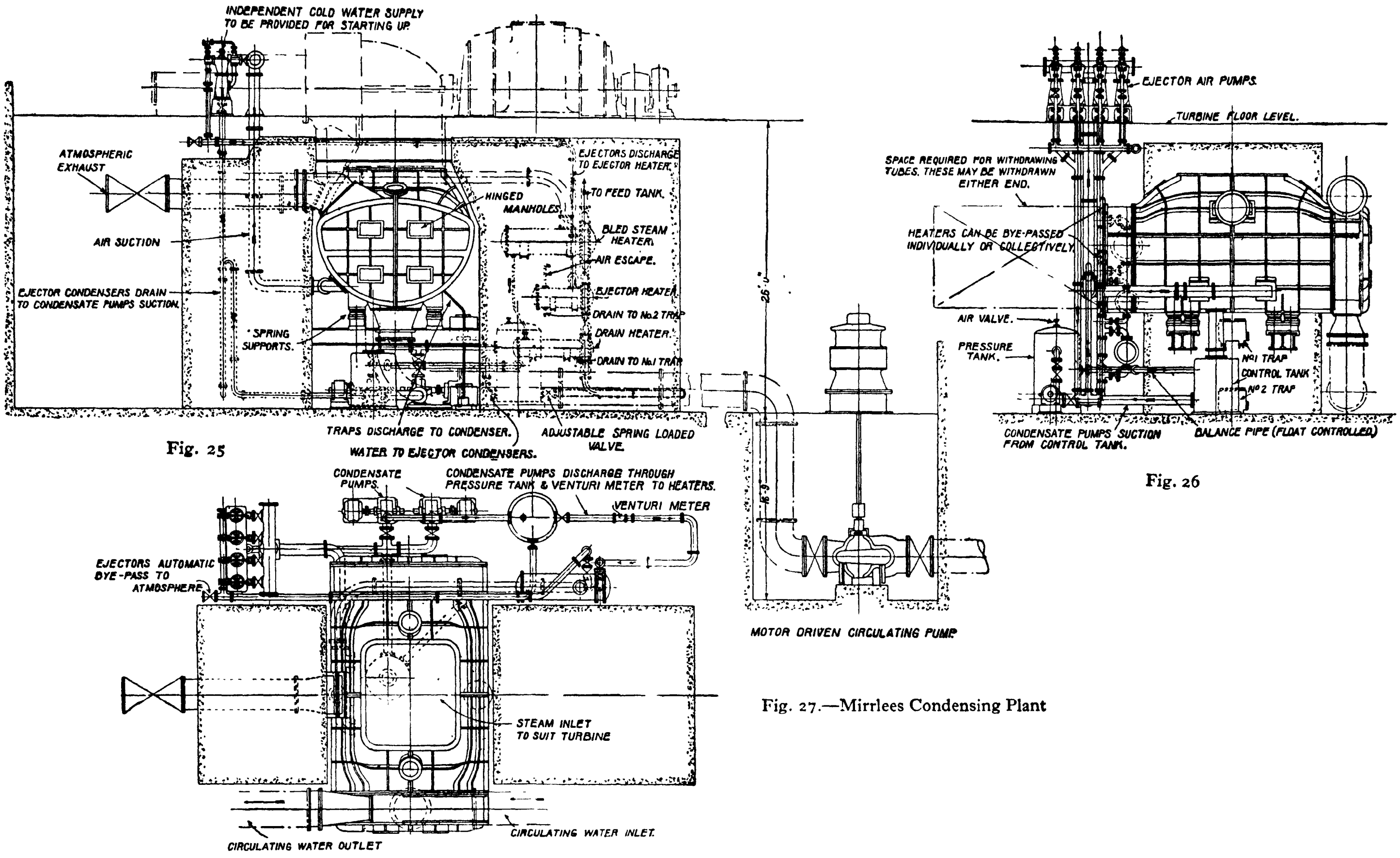

INDEPENDENT COLD WATER SUPPLY TO BE PROVIDED FOR STARTING UP.
ATMOSPHERIC EXHAUST
AIR SUCTION
EJECTOR CONDENSERS DRAIN TO CONDENSATE PUMPS SUCTION.
SPRING SUPPORTS.
HINGED MANHOLES.
EJECTORS DISCHARGE TO EJECTOR HEATER.
TO FEED TANK.
BLED STEAM HEATER.
AIR ESCAPE.
EJECTOR HEATER.
DRAIN TO No. 2 TRAP
DRAIN HEATER.
DRAIN TO No. 1 TRAP
TRAPS DISCHARGE TO CONDENSER.
WATER TO EJECTOR CONDENSERS.
ADJUSTABLE SPRING LOADED VALVE.
26' 0"
6' 9"
MOTOR DRIVEN CIRCULATING PUMP
EJECTOR AIR PUMPS.
TURBINE FLOOR LEVEL.
SPACE REQUIRED FOR WITHDRAWING TUBES. THESE MAY BE WITHDRAWN EITHER END.
HEATERS CAN BE BYE-PASSED INDIVIDUALLY OR COLLECTIVELY.
AIR VALVE.
PRESSURE TANK.
No 1 TRAP
CONTROL TANK
No 2 TRAP
CONDENSATE PUMPS SUCTION FROM CONTROL TANK.
BALANCE PIPE (FLOAT CONTROLLED)
CONDENSATE PUMPS.
CONDENSATE PUMPS DISCHARGE THROUGH PRESSURE TANK & VENTURI METER TO HEATERS.
VENTURI METER
EJECTORS AUTOMATIC BYE-PASS TO ATMOSPHERE.
STEAM INLET TO SUIT TURBINE
CIRCULATING WATER OUTLET
CIRCULATING WATER INLET.

Fig. 25

Fig. 26

Fig. 27.—Mirrlees Condensing Plant

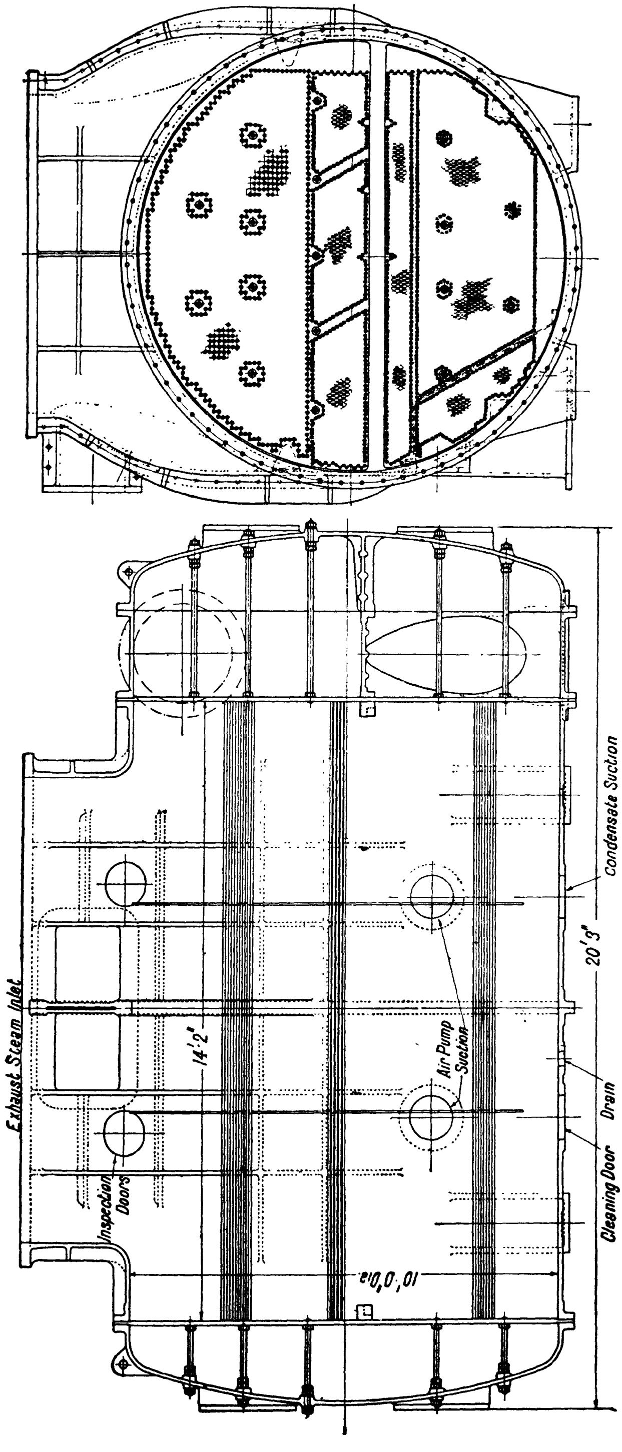


Fig. 28.—Longitudinal and Cross Section of Mirrlees Watson Surface Condenser

per hour; vacuum, 29·1 in. Hg; and inlet temperature of cooling water, 50° F.

The condenser shell is of cast iron suitably ribbed to withstand the vacuum. The tube plates are of rolled brass secured to the shell by a number of collar bolts, thus making a permanent vacuum joint when the end covers are removed for cleaning or inspection purposes. The tube supporting plates are spaced so that the unsupported length of tube does not exceed 100 times the tube diameter, and the tubes are secured to the tube plates by the usual ferrule, with rounded entrance to reduce the entrance losses of the cooling water to a minimum.

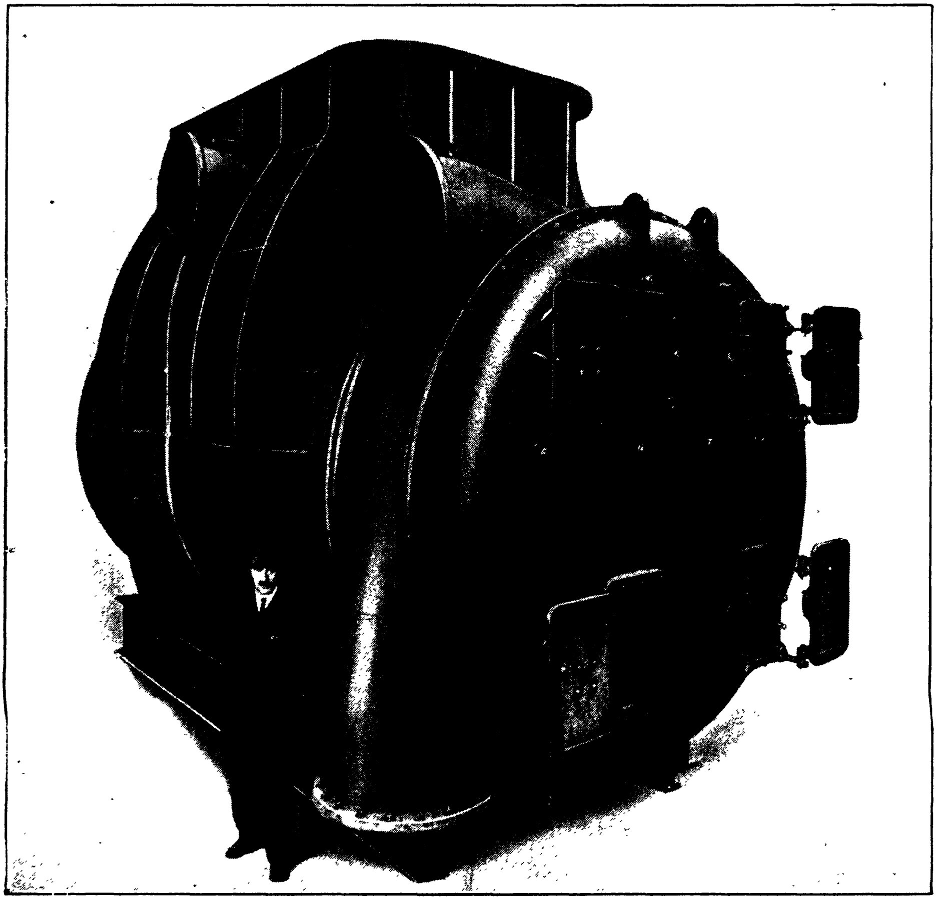


Fig. 29.—Mirrlees Watson Surface Condenser

Fig. 28 shows an example of a round condenser as designed and manufactured by Messrs. The Mirrlees Watson Company for the Glasgow Corporation Electricity Works, Pinkston Power Station.

This condenser works in conjunction with a 10,000-kw. turbo-alternator, and is designed for a normal steam duty of 125,000 lb. of exhaust steam per hour, and maintaining a vacuum of 28·3 in. Hg, when supplied with 19,200 gall. of cooling water per minute at a temperature of 75° F. It is also designed to cope with an overload of 185,000 lb. of exhaust steam per hour, and maintain a vacuum of 27·8 in. of mercury. The cooling surface is 19,000 sq. ft., and the tubes are $\frac{3}{4}$ in. external diameter, 18 S.W.G. thick, and have a composition of 70 per cent copper, 29 per cent zinc, and 1 per cent tin. The water velocity through the tubes is 6·1 ft. per second.

The internal diameter of the shell is 10 ft., and the distance between the tube plates is 13 ft. 7 in. The shell is of cast iron suitably ribbed and stiffened to withstand the working conditions, and the tube plates are supported as shown by the gun-metal tension stays which extend through the water boxes to the end covers.

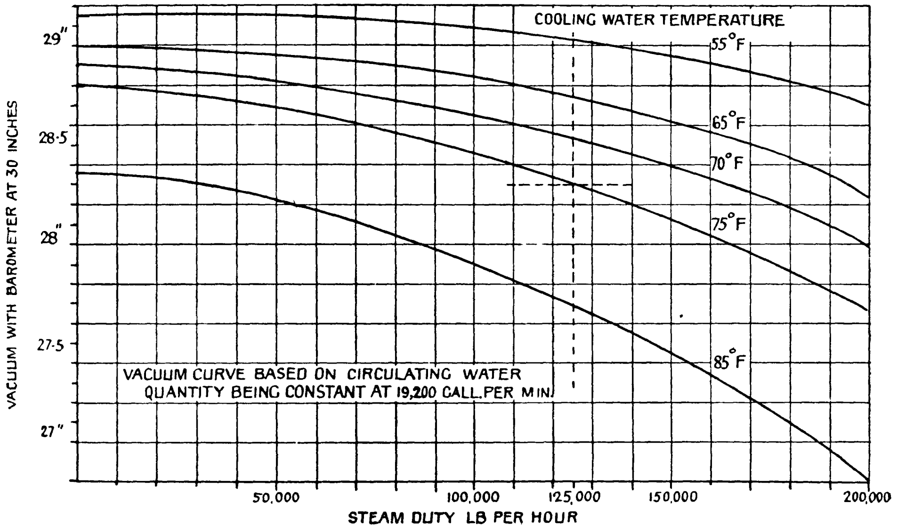


Fig. 30

The exhaust inlet is hooded to the full diameter of the condenser to present a large tube surface to the entering steam, and the tubes in the top third of the tube plate are given a parallel or square pitching to offer as little resistance as possible to the entering steam, and ensure its penetration to the centre of the condenser, so that the whole tube surface is effective. The tube pitching is progressively reduced from top to bottom of the condenser.

An air cooler or devaporizer is arranged for by sheltering a number of tubes under a galvanized baffle as shown. This baffle extends the whole length of the condenser, and is jointed to the condenser shell top and bottom. Adjacent to the tube plates, openings are provided, and the air and vapour entering the openings flow in a more or less horizontal path to the air pump suction branches situated along the condenser shell.

Fig. 29 shows a photograph of this condenser, which gives a good idea of the general construction; it also shows the hinged cleaning doors provided on the end covers.

Fig. 30 shows the guarantee performance curves for this condenser, with varying steam duties and inlet water temperatures. It is customary for the manufacturers to supply performance curves for their condensers.

Table IV gives the results of a test taken on this condenser on 17th September, 1923, and the author is indebted to Messrs. The Mirrlees Watson Company for permission to publish these.

TABLE IV

PINKSTON NO. 4, 10,000-KW. CONDENSING PLANT

Time.	Vacuum in Inches Hg, Corrected to 30 in. Bar.		Vacuum Drop Across Condenser in Inches Hg.	Temperature Corresponding to Vacuum at Turbine Exhaust.	Temperature of Condensate.	Temperature Drop Across Condenser.	Circulating Water.		Lea Recorder, Pounds per Hour.
	Turbine Exhaust.	Air Suction.					Inlet	Outlet.	
p.m.				°F.	°F.	°F.	°F.	°F.	
1·00	28·8	29·0	0·2	84·6	80	4·6	66	78	120,000
1·15	28·8	29·0	0·2	84·6	80	4·6	66	77	112,000
1·30	28·8	29·0	0·2	84·6	79	5·6	66	77	110,000
1·45	28·8	29·0	0·2	84·6	78	6·6	66	76	108,000
2·00	28·85	29·0	0·15	83·3	78	5·3	66	76	104,000
2·15	28·85	28·95	0·1	83·3	79	4·3	66	76	104,000
2·30	28·85	29·0	0·05	83·3	78	5·3	66	76	102,000
2·45	28·85	28·95	0·1	83·3	78	5·3	66	76	98,000
3·00	28·85	28·95	0·1	83·3	78	5·3	66	76	98,000
3·15	28·9	28·95	0·05	82·0	78	4·0	66	76	96,000
3·45	28·9	29·0	0·1	82·0	78	4·0	66	75	92,000
4·00	28·9	29·0	0·1	82·0	78	4·0	66	75	92,000
4·30	28·85	28·95	0·1	83·3	78	5·3	66	76	105,000
4·45	28·85	28·95	0·05	83·3	79	4·3	66	77	112,000
5·00	28·8	28·95	0·1	84·6	80	4·6	66	76	112,000

The following simple analysis of these results may be instructive.

Taking the first reading which is the nearest to the normal full load conditions, we note that the vacuum recorded is 28·8 in. Hg, the inlet water temperature is 66° F., and from fig. 30 the guaranteed vacuum for this temperature is 28·63 in. Hg. The attained vacuum therefore is 0·17 in. Hg higher than the guaranteed vacuum. The rise in temperature of the circulating water is 78 — 66 = 12° F., and assuming that the exhaust steam contains 1000 B.Th.U. per pound, the quantity of circulating water will be

$$\begin{aligned} G &= (120{,}000 \times 1000)/(12 \times 600) \\ &= 16{,}666 \text{ gall. per minute.} \end{aligned}$$

The designed quantity was 19,200 gall. per minute, so that the vacuum performance is better than that shown above, because of the reduced quantity of circulating water. The total vacuum loss is 0·2 in. Hg, and the depression of the condensate temperature is 4·6° F.; this depression of condensate temperature is a direct loss, and 4·6° F. represents a loss of approximately

$$4{\cdot}6/1000 = 0{\cdot}46 \text{ per cent}$$

of the steam consumption of the main turbine, or

$$(120{,}000 \times 0{\cdot}46)/100 = 550 \text{ lb. of steam per hour.}$$

Taking the boiler as giving 6 lb. of steam per pound of coal, and coal at 18*s*. per ton, the above loss represents an annual value of

$$(550 \times 24 \times 7 \times 52 \times 18)/(6 \times 2240 \times 20) = £322.$$

This is quite a moderate loss for a surface condenser from the above cause, but it indicates the importance of keeping the temperature drop across the condenser down to the lowest limits.

On p. 47 an expression was given for the coefficient of performance of a condenser. Working this out for this condenser we get:

$$\begin{aligned}\text{coefficient of performance} &= (T_2 - T_1)/(T_s - T_c + 10)\\ &= (78 - 66)/(84{\cdot}6 - 80 + 10)\\ &= 0{\cdot}822.\end{aligned}$$

This is quite a good value.

The average value of the resistance to heat flow R_a (see p. 10) may be worked out as follows:

$$\begin{aligned}w &= \theta_m/(R_a + R_t + R_w)\\ &= 120{,}000/19{,}000\\ &= 6{\cdot}32 \text{ lb. per square foot.}\end{aligned}$$

And the mean temperature difference θ_m is:

$$\begin{aligned}\theta_m &= [(80 - 66) - (84{\cdot}6 - 78)]/[2{\cdot}3 \times \log_{10}(80 - 66)/(84{\cdot}6 - 78)]\\ &= 9{\cdot}85^\circ \text{ F.}\end{aligned}$$

$$\begin{aligned}\text{Now} \qquad R_w &= C/v^{0{\cdot}82}.\\ \text{From fig. 5} \qquad C &= 2{\cdot}85,\\ \text{and the tube velocity} \quad v &= (6{\cdot}1 \times 16{,}666)/19{,}200\\ &= 5{\cdot}3 \text{ ft. per second.}\\ \therefore\ R_w &= 2{\cdot}85/5{\cdot}3^{0{\cdot}82}\\ &= 0{\cdot}726;\\ \text{and} \qquad R_t &= 0{\cdot}08.\end{aligned}$$

Then substituting the values in the equation, $w = \theta_m/(R_a + R_t + R_w)$, we get:

$$\begin{aligned}6{\cdot}32 &= 9{\cdot}85/(R_a + 0{\cdot}08 + 0{\cdot}726);\\ \therefore\ R_a &= 0{\cdot}75.\end{aligned}$$

CONTRA-FLOW SURFACE CONDENSER

The arrangement of the contra-flow surface condenser for land power stations is as shown in fig. 31. This condenser has a cooling surface of 40,000 sq. ft. and is of the two-flow type with the first flow A nested inside the second or return flow B. The sides of the shell are carried well out beyond the tubes, and are arranged to give a diminishing area from the top to the bottom of the condenser. This arrangement gives a large entrance area for the steam, so that the loss in vacuum is reduced to the minimum. Guide plates C are fitted in the first flow and are arranged as shown so that the area for steam flow is gradually diminished, thus maintaining the steam velocity as condensation proceeds. These baffles also catch the water falling from the upper tubes, and divert it to the condensate outlet. The water is thus prevented from falling on the lower and colder tubes. This arrangement it is claimed gives a condensate temperature commensurate with the vacuum. Devaporizing and air-cooling chambers are arranged beneath the baffles D, and two air outlets are arranged for.

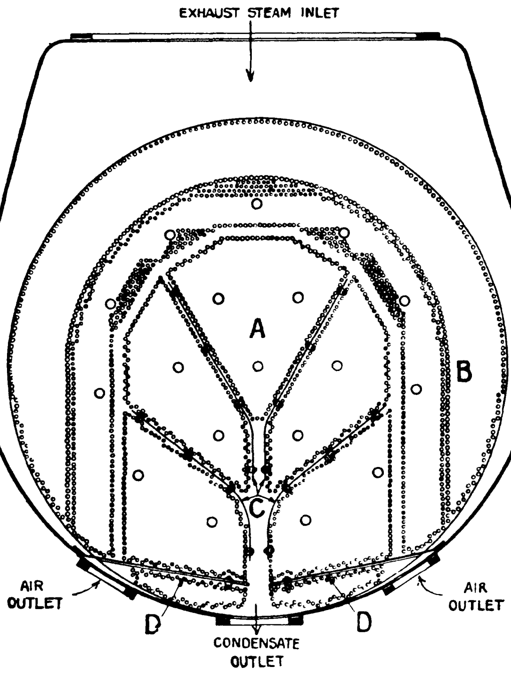


Fig. 31.—Section through Contra-flow Condenser

REGENERATIVE SURFACE CONDENSERS

On p. 71 mention was made of the depression of the condensate temperature and of the loss involved by such depression. This loss varies with the general design of the condenser and must be reduced to a minimum. The regenerative condenser has been designed with this end in view.

It can readily be realized that in a normal condenser the steam which is condensed in the top of the condenser is progressively cooled in its descent to the bottom of the condenser, the cooling being due to the combined effect of the increasing partial air pressure and the contact of the condensed steam on the surface of the cold tubes. The result of these effects is the depression of the condensate temperature below the vacuum temperature at the top of the condenser.

All regenerative condensers are designed on the principle that exhaust steam must penetrate unimpeded to the bottom of the condenser. The relatively cold condensate, therefore, in falling from the bottom row of tubes falls through the relatively hot steam which has penetrated to the bottom of the condenser. The condensate is thereby reheated by direct contact to a temperature approximating that of the steam.

There have recently been placed on the market three condensers designed on this principle; these are the radial flow condenser of the Westinghouse Company of America; the regenerative condenser of Messrs. G. & J. Weir, Ltd.; and that designed by Messrs. Brown Boveri of Switzerland.

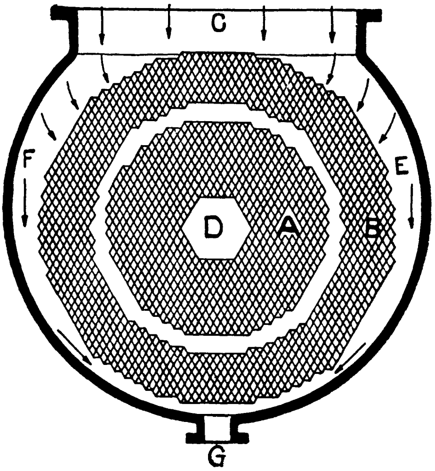


Fig. 32.—The Westinghouse Regenerative Condenser

Fig. 32 illustrates the radial flow condenser of the Westinghouse Company.

The tubes are arranged in two concentric flows A and B: the inner flow A being the first flow, and the outer flow B being the second or return flow. The exhaust steam enters at C and, entering the tubes radially, flows to the air suction D arranged in the centre. Two belts E and F are formed round the sides, so that the steam can penetrate right to the bottom or condensate outlet. The condensate, therefore, in falling from the bottom tubes, is thereby raised in temperature and leaves the condenser by the outlet branch G at approximately steam temperature. Since the first flow A is arranged inside the second flow B a large air-cooling effect is obtained, and the air leaving by the air outlet D is devaporized of much of its associated vapour, and the air considerably cooled. A feature of this design is the short penetration of the steam into the tube bank to the air suction.

Fig. 33 illustrates the regenerative condenser of Messrs. G. & J. Weir, Ltd.

This condenser has a large clear passage E arranged down the centre, so that exhaust steam can penetrate to the bottom and thereby reheat the condensate to approximately steam temperature. A two-flow passage for the cooling water is adopted, these flows being arranged in pairs on opposite sides of the passage E. The first flow is A, and the second flow is B. Twin inlet and outlet cooling water branches H and J respectively are employed, and the arrangement lends itself admirably to the modern divided condenser, so that one-half can be cleaned while the other half is in commission. Baffles K are arranged so that a very large and efficient air cooler is provided, the air suction branches being at D.

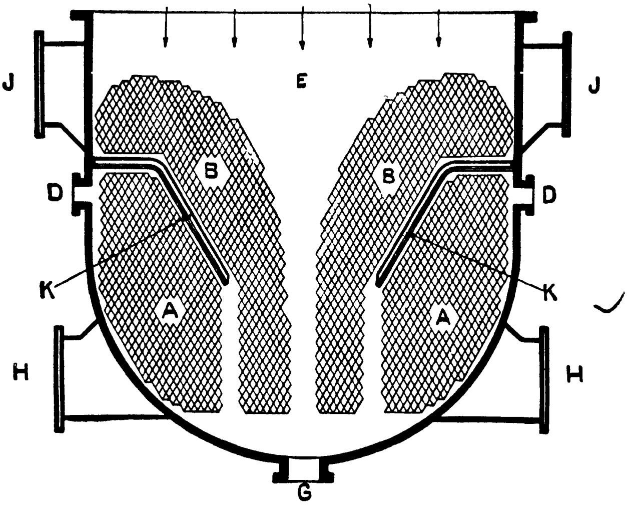


Fig. 33.—The Weir Regenerative Condenser

Fig. 34 illustrates the design adopted by Messrs. Brown Boveri.

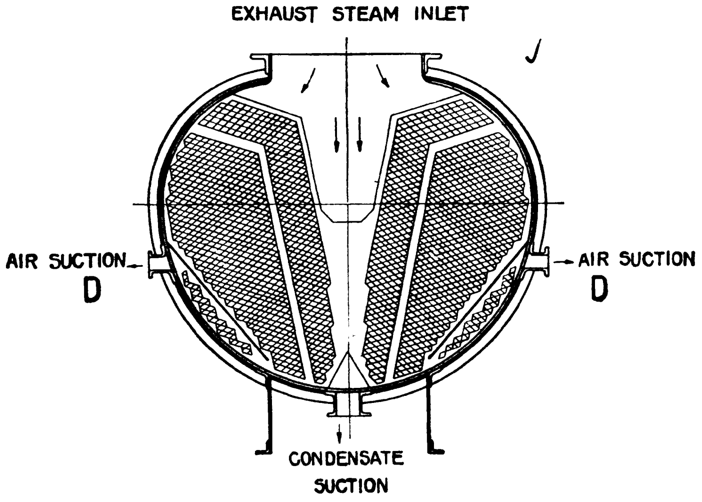


Fig. 34.—The Brown-Boveri Regenerative Condenser

This arrangement is very similar to that adopted by Messrs. G. & J. Weir, Ltd., the main difference being in the perforated baffle plate arranged over the air suctions D.

JOHN CRANE METALLIC PACKING FOR CONDENSER TUBES

A recent development in condenser tube packing is that known as the John Crane Metallic Packing. This packing was first introduced in the United States, and is now manufactured in England by Crane Packing, Ltd., Slough, England, to whom the author is indebted for the following illustration.

The ideal condenser tube packing should be one which effectively prevents leakage of the circulating water into the vacuum or steam space under varying temperature conditions; the pressure of the packing on the tube should not be such as to cause crushing or deformation of the tube. The tube should be free to expand and contract longitudinally, and the act of packing should be simple and easily performed.

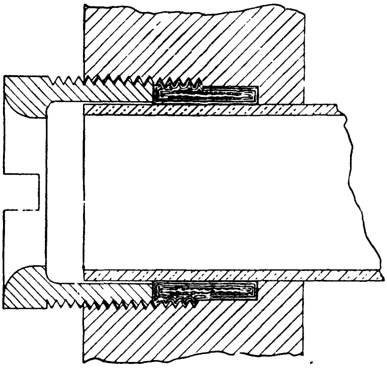


Fig. 35.—" John Crane " Packing for Condenser Tubes

Much has been written regarding the disintegration of condenser tubes, and two schools have developed from the various discussions which have taken place. One school has advanced the theory that condenser tube corrosion is purely galvanic or electrolytic in action. The other school holds that the corrosion is due to metallic impurities or improper selection of the materials used in the tube manufacture, the effect being attributed to the action known as local action. Both schools are probably so far correct, and the corrosion which is commonly experienced may be due to both causes. It is now generally accepted that the materials used in the manufacture of condenser tubes should be the purest obtainable and, especially for condensers using sewage or sea water for cooling, some form of protection against galvanic action should be provided. This protection is provided in the Crane process by employing a metallic packing. This is made up of thin foil of Babbit metal built up of many spiral layers. Entrapped between these layers of Babbit metal is a length of fabric tape or other suitable expansion material.

These tape entrapped Babbit foil rings are flexible and compressible, and when caulked in the stuffing box in the tube plate expand and form a metallic contact between the tube and the tube plate. This metallic contact is claimed by the Crane Company to greatly minimize galvanic or electrolytic action. The assembly of this packing is shown in fig. 35.

This " John Crane " flexible metallic packer, as it is called, is a one-piece unit and is complete in itself. It is slipped into position over the tube and into the stuffing box, and caulked up with a suitable caulking tool, using a few light hammer blows. When the condenser is thus packed the hydrostatic test up to 30 lb. per square inch can be put on before the ferrules are screwed into the stuffing box. When the hydrostatic test is complete the ferrules are screwed in hard against the metallic packers. The number of stuffing boxes that can be packed per hour by each man is approximately 250. Where extra deep stuffing boxes have to be dealt with, a fibre or other suitable spacer ring can be inserted first in the bottom of the box, and the John Crane metallic packer caulked in on the top of this ring. It is necessary to space the ferrule correctly with regard to the clearance between the end of the tube and the shoulder of the ferrule.

CHAPTER II

Low-level Jet Condensers

A jet condenser is a much simpler and a less costly piece of apparatus than a surface condenser, and would be installed frequently were it not for the fact that the condensed steam, which is pure distilled water and when properly treated an ideal boiler feed, is thrown to waste in the cooling water. The jet condenser should only be installed when the cooling water itself is, or can be, easily and cheaply made suitable for boiler feed, or when a cheap source of boiler feed water is available.

In jet condensers the steam to be condensed and the cooling water are intimately mixed by the breaking up of the water into a rain or spray, and allowing this rain to fall down through the body of the steam, or by discharging the water through suitably shaped nozzles into the body of the steam. These nozzles atomize the water by the sudden release of pressure and give a finely divided spray, thus presenting a large water surface to the steam and ensuring a rapid and complete condensation.

The quantity of the cooling water depends on its temperature, on the mass and temperature of the steam to be condensed, and the temperature of the hotwell. Since the supply of water to the boiler (feed water) is usually taken from the hotwell, it is obviously an advantage for the hotwell to be as warm as possible. The cooling water used should only be just sufficient to produce the desired vacuum.

Some makers give a contra-flow to the steam and water; the steam enters at the bottom and flows upwards, while the water enters at the top and falls downwards; the air pump connection is at the top where the air is coldest. Other makers give a parallel flow to the steam and water by discharging the cooling water downwards through nozzles into the body of the steam; the steam and water entering at the top and passing to the bottom where the mixture is withdrawn by an extraction pump, a special compartment being provided for the cooling of the air.

The parallel-flow arrangement is generally best suited for turbine work where the exhaust comes from the underside of the turbine.

The quantity of condensing water required is given by equation (16) p. 16, viz.,

$$G = [W(qL + t_1 - t_2)]/[600(T_2 - T_1)].$$

Since, in a jet condenser, the steam and cooling water mix together the temperature of the cooling water will be the temperature of the condensate, i.e. $t_2 = T_2$,

$$\therefore\ G = [W(qL + t_1 - T_2)]/[600(T_2 - T_1)]. \ldots\ldots (37)$$

It is customary practice to have a small temperature difference between the outgoing water and the incoming steam of 2° to 4° F., that is, the value of $(t_1 - T_2)$ in equation (37) is 2° to 4° F.

The cause of this temperature difference is the presence of air, which forms a blanket on the surface of the water drops as the steam condenses in a manner similar to that by which the air blanket is formed round the tubes of a surface condenser. With a jet condenser arranged contra-flow or with an air cooler it is possible, however, to attain a temperature difference approaching zero. With the contra-flow type of condenser there is a great air cooling effect, due to the air passing upwards and being progressively cooled by the falling condensing water which is being progressively heated; the net result of the arrangement being that the air leaves the condenser at a much lower temperature than the vacuum temperature, so that the air pump can effectively deal with the air and maintain the partial air pressure at the water outlet approximately zero.

To get the same results with an air pump only, i.e. without this air cooling effect, the cost and size of the pump would be prohibitive.

Allowing then a temperature difference of 2° F. for a contra-flow condenser, and taking the heat contents of the steam as 1000 B.Th.U., equation (37) becomes:

$$G = (1002W)/[600(T_2 - T_1)]. \ldots\ldots (38)$$

The quantity of condensing water G, as found from equation (37), makes no allowance for the air cooling and vapour condensing effect, and to the above quantity must be added a further quantity which may be found thus:

Let t_a be the temperature of the air leaving the condenser in degrees F.;
G_a, the quantity of condensing water necessary for air cooling and vapour condensing (gallons per minute);
L, the latent heat of the vapour at temperature T_2;
γ, the specific heat of the air $= 0{\cdot}2375$; and
m, the mass of air being dealt with in pounds per hour.

The problem here is the same as that for an air cooler in a surface condenser, so that the mass of vapour condensed out will be given by equation (32), p. 42:

$$M_s = (V_{1a}/V_{1s} - V_{2a}/V_{2s})m.$$

The heat given up by this mass of vapour condensing out and the air in cooling will be found from equation (33), p. 42:

$$H = m[L(V_{1a}/V_{1s} - V_{2a}/V_{2s}) + \gamma(T_2 - t_a)],$$

$(T_2 - t_a)$ being the fall in temperature of the air.

The heat taken up by the water is

$$H = 600G_a(T_2 - T_1).$$

Equating these equal quantities of heat we get:

$$G_a = [m\{L(V_{1a}/V_{1s} - V_{2a}/V_{2s}) + \gamma(T_2 - t_a)\}]/[600(T_2 - T_1)]. \,..(39)$$

The total quantity of cooling water necessary then is $G + G_a$.

Example.—A jet condenser of the contra-flow type is to condense 76,000 lb. of steam per hour at a vacuum of 28 in. Hg. The inlet temperature of the condensing water is 60° F., and from considerations of air pump capacity the temperature of the air and vapour leaving the condenser must not exceed 65° F. The air to be extracted is 60 lb. per hour. Assume the vacuum at the air-suction branch to be 28·1 in. Hg. Calculate the total quantity of condensing water required.

From equation (38) the condensing water necessary for the condensation of the steam only is

$$G = (1002W)/[600(T_2 - T_1)].$$

Now T_2 is taken 2° F. lower than the vacuum temperature, i.e.

$$\begin{aligned} T_2 &= 101 - 2 \\ &= 99^\circ \text{ F.} \\ \therefore\ G &= (76{,}000 \times 1002)/[600(99 - 60)] \\ &= 3253 \text{ gall. per minute.} \end{aligned}$$

Before equation (39) can be solved, V_{1a} and V_{2a} must first be found. As before, we have from equation (28),

$$\begin{aligned} V_{1a} &= [30 \times 12{\cdot}39(459{\cdot}6 + 99)]/[(28{\cdot}13 - 28)(459{\cdot}6 + 32)] \\ &= 3248 \text{ c. ft. per pound of air.} \\ V_{2a} &= [30 \times 12{\cdot}39(459{\cdot}6 + 65)]/[(29{\cdot}38 - 28{\cdot}1)(459{\cdot}6 + 32)] \\ &= 310 \text{ c. ft. per pound.} \end{aligned}$$

From the steam tables:

$$\begin{aligned} V_{1s} &= 361 \text{ c. ft. per pound,} \\ V_{2s} &= 1024 \text{ c. ft. per pound,} \\ \text{and } L &= 1036{\cdot}2. \end{aligned}$$

Then substituting these values in equation (39) we get:

$$\begin{aligned} G_a &= [60\{1036{\cdot}2(3140/361 - 309/1024) + 0{\cdot}2375(99 - 65)\}]/[600(99 - 60)] \\ &= [60(9009 + 8{\cdot}07)]/(600 \times 39) \\ &= 23{\cdot}12 \text{ gall. per minute.} \end{aligned}$$

The total quantity of condensing water therefore will be

$$\begin{aligned} &= 3253 + 23{\cdot}12 \\ &= 3276{\cdot}12 \text{ gall. per minute,} \\ &= \text{say, } 3276 \text{ gall. per minute.} \end{aligned}$$

Example.—The following figures were taken during the test of a contraflow jet condenser. Find the mass of air and vapour being dealt with by the air pump.

Quantity of condensing water entering condenser, 2500 gall. per minute.
Vacuum in exhaust branch, 27·5 in. Hg.
Temperature of condensate, 104° F.
Inlet temperature of condensing water, 65° F.
Temperature of air suction branch, 75° F.
Vacuum at air suction branch, 27·64 in. Hg.
Quantity of steam condensed, 58,120 lb. per hour.

From equations (38) and (39):

$$\begin{aligned}(G + G_a) = {} & [1002W]/[600(T_2 - T_1)] \\ & + [m\{L(V_{1a}/V_{1s} - V_{2a}/V_{2s}) + \gamma(T_2 - t_a)\}]/[600(T_2 - T_1)].\end{aligned}$$

As before we have:

$$\begin{aligned}V_{1a} &= [(30 \times 12{\cdot}39)(459{\cdot}6 + 104)]/[(27{\cdot}81 - 27{\cdot}5)(459{\cdot}6 + 32)] \\ &= 1375 \text{ c. ft. per pound of air;} \\ V_{2a} &= [(30 \times 12{\cdot}39)(459{\cdot}6 + 75)]/[(29{\cdot}13 - 27{\cdot}64)(459{\cdot}6 + 32)] \\ &= 271 \text{ c. ft. per pound of air;} \\ (G + G_a) &= 2500 \text{ gall. per minute;} \\ T_1 &= 65^\circ \text{ F;} \\ T_2 &= 104^\circ \text{ F;} \\ t_a &= 75^\circ \text{ F;} \\ L &= 1033{\cdot}4 \text{ B.Th.U.;} \\ W &= 58{,}120 \text{ lb. per hour;} \\ V_{1s} &= 313{\cdot}3 \text{ c. ft. per pound;} \\ V_{2s} &= 743 \text{ c. ft. per pound.}\end{aligned}$$

Substituting these values in the above equation we get:

$$\begin{aligned}2500 = {} & [58{,}120 \times 1002]/[600(104 - 65)] \\ & + [m\{1033{\cdot}4(1375/313{\cdot}3 - 271/743) + 0{\cdot}2375(104 - 75)\}]/[600(104 - 65)]; \\ 58{,}500{,}000 &= 58{,}236{,}240 + 4157m; \\ \text{or} \quad m &= 263{,}760/4157 \\ &= 63{\cdot}4 \text{ lb. of air per hour.}\end{aligned}$$

The mass of vapour being dealt with by the air pump can now be calculated from equation (32):

$$\begin{aligned}M_s &= (V_{1a}/V_{1s} - V_{2a}/V_{2s})m \\ &= (1375/313{\cdot}3 - 271/743)63{\cdot}4 \\ &= 255{\cdot}1 \text{ lb. of vapour per hour.}\end{aligned}$$

The total mass of air and vapour being dealt with by the air pump will then be

$$\begin{aligned}&= 255{\cdot}1 + 63{\cdot}4 \\ &= 318{\cdot}5 \text{ lb. per hour.}\end{aligned}$$

In jet condensers the mass of air (*m*) which has to be extracted from the condenser may be divided into three parts: one part being the air which leaks into the vacuum system, a second part the air which comes over with the steam, and the third part the air which comes in with the condensing water. As is well known, water which has been exposed to the atmosphere contains air and small quantities of other gases in solution. The quantity of air so held in solution depends upon the treatment the water has received while exposed to the atmosphere. For instance, if water be taken from a mountain stream and the dissolved air in it be measured, the quantity will be found to be about 3 per cent by volume. If, however, water be taken from the open hotwell of a steam condensing plant, the quantity of dissolved air will be found to be about 0·5 per cent by volume. These are extreme cases, and in general it will be found that the colder the water and the greater the agitation to which the water has been subjected, the greater will be the absorption of air up to a limit of about 3 per cent by volume.

The cooling water, then, supplied to a jet condenser contains normally a large percentage of dissolved air, and under the combined influence of the vacuum, the breaking up of the water, and the heating which takes place in the condenser this air is liberated and has to be dealt with by the air pump.

When a jet condensing plant is installed, the boiler is usually fed with raw water, and the air held in solution in the boiler feed ultimately comes over with the steam to the condenser.

DIAMETER OF THE COOLING WATER INLET PIPE FOR A CONTRA-FLOW JET CONDENSER

With the low-level jet condenser it is customary practice to rely on the vacuum drawing cooling water into the condenser from the river or pond. The necessary area of the cooling water inlet pipe is governed by the head against which the water has to be lifted, the vacuum, and the loss of head due to friction in the length of the cooling water inlet pipe.

Let the following diagram (fig. 36) illustrate our condenser.

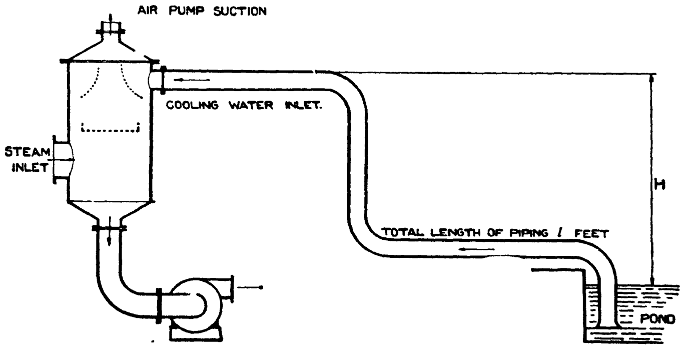


Fig. 36.—Low-level Jet Condenser

Let H be the head in feet from the water-level in the pond or river to the top of the water inlet branch on the condenser;
p, the absolute pressure in the condenser in inches of mercury, i.e. 30 in. minus the vacuum;
l, the total length of the piping in feet;
n, the number of right-angled bends;
D, the internal diameter of the pipe in feet;
v, the water velocity in the pipe in feet per second;
G, the quantity of cooling water passing in gallons per minute;
F, the total friction loss in the piping; and
H_1, the head causing flow at the inlet branch.

Then the head causing flow at the inlet branch into the condenser will be the head equivalent to the condenser vacuum less the lift H and the friction loss,

i.e. $$H_1 = [(30 - p)14{\cdot}7 \times 2{\cdot}3]/30 - H - F.$$

The friction F is made up of the friction losses at the entrance to the pipe and in the bends. The entrance loss may be neglected if the end of the pipe is bell-mouthed; if a strainer is fitted to the end of the pipe the friction loss across the strainer should be added.

For practical purposes the losses at the bends may be computed by allowing for every easy right-angled bend a length of straight pipe of 10 to 15 diameters, and for every sharp right-angled bend 30 to 36 diameters. These lengths are in agreement with the results of a number of experiments carried out at the Yorkshire College. (See *Engineering*, 25th September, 1896, p. 390.)

The friction in the straight lengths of the pipe may be found from the usual formula:

$$H_p = (flv^2)/(2gm),$$

where f is a constant depending on the roughness of the inside of the pipe, and for ordinary cast-iron piping may be taken at 0·01; and m is the hydraulic mean depth which, for a circular pipe, is D/4.

The head H_1 causing flow will then be, for a bell-mouth inlet with no strainer,

$$H_1 = 1{\cdot}127(30 - p) - H - [(4fv^2)\{l + (15nD)\}]/[2gD] \ldots . (40)$$

The velocity of flow v will be that due to the effective head H_1, and is given by the relation

$$v = \sqrt{2gH_1} \text{ ft. per second.}$$

The quantity of water flowing will be

$$Q = (\pi D^2 v)/4 \text{ c. ft. per second, or, in gallons per minute,}$$

$$G = (\pi D^2 v 60)/(4 \times 6{\cdot}23)$$
$$= (\pi D^2 v)/0{\cdot}415.$$

Example 1.—Find the quantity of cooling water which will be drawn into a contra-flow jet condenser by the vacuum from a pond situated 100 yd. from the condenser through a 6-in. bore pipe. The vertical distance from the water-level in the pond to the top of the cooling water inlet branch on the condenser is 12 ft., and the vacuum in the condenser is 26 in. Hg with a 30 in. Hg barometer. There are five right-angled bends in the cooling water pipe.

From equation (40) the head H_1 causing flow will be:

$$
\begin{aligned}
H_1 &= 1\cdot127(30 - 4) - 12 - \{v^2/(2g)\}[(4 \times 0\cdot01/0\cdot5)(312 + 5 \times 15 \times 0\cdot5)] \\
&= 29\cdot3 - 12 - 27\cdot96 \times v^2/(2g) \\
&= 17\cdot3 - 27\cdot96 \times v^2/(2g).
\end{aligned}
$$

$$
\begin{aligned}
\text{Now} \qquad H_1 &= v^2/(2g), \\
\text{and} \qquad H_1 &= 17\cdot3 - 27\cdot96v^2/(2g). \\
\therefore\ v^2/(2g) &= 17\cdot3 - 27\cdot96v^2/(2g), \\
\text{i.e. } v &= \sqrt{(17\cdot3 \times 2g)/28\cdot96} \\
&= 6\cdot18 \text{ ft. per second.} \\
\text{Then} \qquad G &= (\pi D^2 v)/0\cdot415 \\
&= (\pi \times 0\cdot25 \times 6\cdot18)/0\cdot415 \\
&= 11\cdot69 \text{ gall. per minute.}
\end{aligned}
$$

Example 2.—In the preceding example find the temperature of the water in the pond which will cause the flow of cooling water to stop, and the vacuum in the condenser when the flow ceased.

If the temperature of the pond gradually rises the vacuum in the condenser will gradually fall, and this fall in vacuum will cause the quantity being drawn in to gradually diminish till a condition is reached such that the vacuum will balance the static head H_1, and the flow into the condenser will cease. The water will momentarily stand up the pipe the height H and be suspended there like a water barometer. The problem then is to find the temperature which corresponds to the water vapour pressure of $(34 - H)$, 34 ft. being taken as the height of the water barometer.

The absolute pressure in pounds per square inch at the top of the water column will be

$$
\begin{aligned}
&= 14\cdot7 - (H \times 14\cdot7)/34 \\
&= 14\cdot7 - (12 \times 14\cdot7)/34 \\
&= 14\cdot7 - 5\cdot19 \\
&= 9\cdot51 \text{ lb. per square inch.}
\end{aligned}
$$

The temperature which corresponds to this absolute pressure of 9·51 lb. per square inch is obtained from the steam tables, namely, 190·54° F.

This is the theoretical temperature, but actually the flow would cease before this temperature was reached, because of the air in the cooling water.

The vacuum in the condenser when the flow ceased would be

$$
\begin{aligned}
&= 30 - (9\cdot51 \times 30)/14\cdot7 \\
&= 30 - 19\cdot4 \\
&= 10\cdot6 \text{ in. of mercury.}
\end{aligned}
$$

HEAD OF WATER IN TRAYS OF CONTRA-FLOW JET CONDENSERS

The contra-flow jet condenser is normally arranged with two or three water trays perforated with holes to break up the water. Sufficient water head must be arranged over the trays to ensure that they are always full.

Let fig. 37 represent such a condenser having two trays. Taking the top tray first:

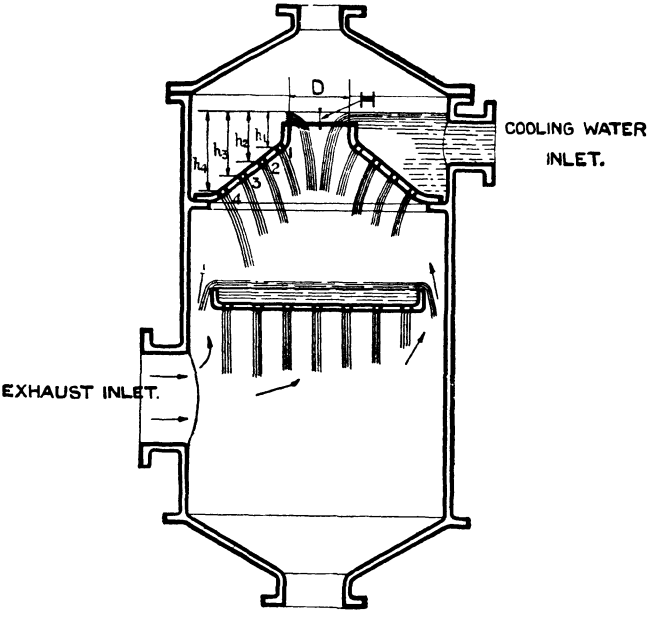


Fig. 37.—Contra-flow Jet Condenser

Let h_1, h_2, &c., be the head over the successive rows of holes in feet.

Let A_1, A_2, &c., be the respective total areas of the successive rows of holes in square feet.

Let G be the quantity of cooling water passing in gallons per minute.

It is customary to allow about 50 per cent of the cooling water to flow over the top of the tray, so that the total quantity of cooling water which flows through the holes will be 0·5G.

At the first row the water velocity leaving the holes will be

$$v_1 = \sqrt{2gh_1} \text{ ft. per second,}$$

and the quantity passing the row will be

$$A_1v_1 = A_1\sqrt{2gh_1} \text{ c. ft. per second,}$$

or, allowing a coefficient of discharge of 0·62,

$$A_1v_1 = 0{\cdot}62\, A_1\sqrt{2gh_1}.$$

Similarly, at the second row we have

$$A_2 v_2 = 0{\cdot}62 A_2 \sqrt{2gh_2},$$

and so on for any number of rows.

The total quantity passing the rows will be:

$$A_1 v_1 + A_2 v_2 + \&c. = 0{\cdot}62\{A_1\sqrt{2gh_1} + A_2\sqrt{2gh_2} + \&c.\},$$

$$\text{i.e. } (0{\cdot}5 \times G \times 10)/(60 \times 62{\cdot}3) = 0{\cdot}62\{A_1\sqrt{2gh_1} + A_2\sqrt{2gh_2} + \&c.\}.$$

$$\therefore G = 463{\cdot}5\{A_1\sqrt{2gh_1} + A_2\sqrt{2gh_2} + \&c.\} \text{ gall. per minute.}$$

If n_1, n_2, &c., be the number of holes in the successive rows, and A the area of each hole, the above equation becomes:

$$G = 463{\cdot}5A\{n_1\sqrt{2gh_1} + n_2\sqrt{2gh_2} + \&c.\} \text{ gall. per minute...(41)}$$

When the design of this tray is laid out on the drawing board, the number of holes in each successive row should be arranged so that the above equation is satisfied.

For the second tray we have, allowing 50 per cent of the water to flow over the top of the tray,

$$G = 463{\cdot}5An\sqrt{2gh} \text{ gall. per minute.}$$

The head of water from the still water-level to the top edge of tray should be from 1½ to 3 in., but not more than 3 in.

The water flowing over the top edge of the tray will take up a free vortex motion partially destroyed by the incoming water from the inlet branch.

If D be the top diameter of the tray in feet, and H the head over the top of the tray in feet, we get

$$(0{\cdot}5G)/(60 \times 6{\cdot}23) = C_d\pi/4 \times D^2 \times \sqrt{2gH},$$

$$\text{i.e.} \quad G = 587C_dD^2\sqrt{2gH} \text{ gall. per minute............(42)}$$

Where C_d is the coefficient of discharge, and varies with the ratio of H to D, the experimental values of C_d for various ratios of H to D are:

H/D	0·2	0·3	0·4	0·5	0·6	0·7
C_d	0·185	0·195	0·21	0·235	0·28	0·35

The diameter of the bottom tray is generally fixed up from considerations of the available space inside the shell, and if 50 per cent of the cooling water is arranged to pass through the bottom holes the head over the top edge of the tray will adjust itself according to the fixed diameter of the tray.

Example.—In the lay-out of a contra-flow jet condenser there are six rows of ½ in. diameter holes at respective heads of 6, 8, 10, 12, 14, and 16 in.

The quantity of cooling water passing is 700 gall. per minute. Arrange the depth of water over the top of the top tray to be 3 in. Find the necessary number of holes in each row so that 50 per cent of the cooling water passes through the holes and the other 50 per cent through the diameter D at the top of the top tray.

From equation (42) we have

$$G = 587C_dD^2\sqrt{2gH},$$
$$\text{i.e. } D = \sqrt{G}/(\sqrt{587C_d} \times \sqrt[4]{2gH}).$$

Provisionally fix the ratio H/D at 0·2, then $C_d = 0{\cdot}185$ (from table);

$$\begin{aligned}\text{and } D &= \sqrt{700}/(\sqrt[4]{64{\cdot}4 \times 3/12} \times \sqrt{587} \times 0{\cdot}185)\\ &= 1{\cdot}268 \text{ ft.}\\ &= 15{\cdot}216 \text{ in.}\\ &= \text{say } 15\tfrac{1}{4} \text{ in.}\end{aligned}$$

The ratio H/D, therefore, is 3/15·25 = 0·19, that provisionally used was 0·2, so that the proper coefficient of discharge will be between 0·195 and 0·185, and may be found by trial and error; the above diameter of 15·25 in. is, however, sufficiently near for practical purposes.

Again, from equation (41)

$$G = 463{\cdot}5A\{n_1\sqrt{2gh_1} + n_2\sqrt{2gh_2} + \&\text{c.}\}.$$

This equation does not solve the problem completely, since n_1, n_2, &c., are unknown. Generally, the quickest way to satisfy the equation is to assume a circumferential pitch of holes, and fix the number of holes in $(n - 1)$ rows, then find the quantity of water passing this number of rows; the total quantity passing, less the quantity passing $(n - 1)$ rows, gives the quantity passing the remaining rows.

Let us fix the number of holes in the top five rows, and let the assumed pitch be 5 in. From the drawing of this tray the pitch circle diameters of the respective rows were 15, 18, 22, 26, 30, and 34 in. The number of holes in the rows, beginning from the top, will therefore be $n_1 = 9$, $n_2 = 11$, $n_3 = 14$, $n_4 = 16$, and $n_5 = 19$.

The quantity g_1 admitted by the first or top row will be:

$$\begin{aligned}g_1 &= \tfrac{1}{2}(463{\cdot}5 \times 0{\cdot}196 \times 8/144) \times \sqrt{h_1}n_1\\ &= 5{\cdot}05 \times 0{\cdot}707 \times 9/2\\ &= 16{\cdot}05 \text{ gall. per minute.}\end{aligned}$$

For the second row:

$$\begin{aligned}g_2 &= \tfrac{1}{2} \times 5{\cdot}05 \times \sqrt{8/12} \times 11\\ &= 5{\cdot}05 \times 0{\cdot}816 \times 11/2\\ &= 22{\cdot}65 \text{ gall. per minute.}\end{aligned}$$

For the third row:

$$\begin{aligned} g_3 &= \tfrac{1}{2} \times 5{\cdot}05 \times \sqrt{10/12} \times 14 \\ &= 5{\cdot}05 \times 0{\cdot}913 \times 14/2 \\ &= 32{\cdot}25 \text{ gall. per minute.} \end{aligned}$$

For the fourth row:

$$\begin{aligned} g_4 &= \tfrac{1}{2} \times 5{\cdot}05 \times \sqrt{12/12} \times 16 \\ &= 40{\cdot}4 \text{ gall. per minute.} \end{aligned}$$

For the fifth row:

$$\begin{aligned} g_5 &= \tfrac{1}{2} \times 5{\cdot}05 \times \sqrt{14/12} \times 19 \\ &= 5{\cdot}05 \times 1{\cdot}08 \times 19/2. \\ &= 51{\cdot}8 \text{ gall. per minute.} \end{aligned}$$

The quantity to be admitted at the sixth row of holes will therefore be:

$$\begin{aligned} g_6 &= 350 - 51{\cdot}8 - 40{\cdot}4 - 32{\cdot}25 - 22{\cdot}65 - 16{\cdot}05 \\ &= 350 - 163{\cdot}15 \\ &= 186{\cdot}85 \text{ gall. per minute.} \end{aligned}$$

The number of holes in the sixth row which will pass this quantity is:

$$\begin{aligned} g_6 &= \tfrac{1}{2} \times 5{\cdot}05\sqrt{16/12} \times n_6, \\ \text{i.e. } n_6 &= (2 \times 186{\cdot}85)/(5{\cdot}05 \times 1{\cdot}154) \\ &= \text{say } 64 \text{ holes.} \end{aligned}$$

This number of holes would give a pitch of 1·81 in. in the sixth row. This pitch is too close, the assumed pitch of 5 in. must therefore be decreased.

NECESSARY VOLUME OF CONTRA-FLOW JET CONDENSER

The volumetric capacity of a contra-flow jet condenser should not be less than a fourth of the volume of the cylinder or cylinders exhausting into it, and need not be more than a half of the volume. If the engine be a very quick running one, a third of the capacity is generally sufficient.

For turbine work or where the steam supply is continuous and not intermittent, like the exhaust from a reciprocating engine, a capacity based on 0·008 to 0·011 c. ft. per pound of steam per hour gives reasonable results. The objection to a large condenser is that it takes a long time to raise the vacuum, and the first cost is high, while the objection to a small condenser is its liability to flooding and overflowing to the engine cylinders. For this reason jet condensers are usually fitted with an automatic vacuum breaker, so arranged that should the water rise dangerously high the vacuum is broken. These vacuum breakers are usually float operated, the float rising with the water and opening a valve which permits atmospheric air to enter the condenser and so break the vacuum.

NECESSARY AREA OF SPRAY NOZZLES IN PARALLEL-FLOW CONDENSER

As was stated on p. 77 the parallel-flow condenser employs spray nozzles in place of perforated trays to break up the cooling water and give good mixing of the steam and water. The pressure drop across these spray nozzles varies with the make of condenser, and should not exceed 6 in. Hg.

If A be the area of each nozzle in square inches;
H_1, the pressure drop in feet, or effective head;
C_d, the coefficient of discharge;
G, the quantity of cooling water in gallons per minute; and
N, the number of nozzles,

we have the following relation:

$$G = 2{\cdot}6NC_dA\sqrt{2gH_1} \text{ gall. per minute} \ldots\ldots\ldots\ldots (43)$$

The value of C_d varies with the design of nozzle, but should not be less than 0·8.

If v_v be the nozzle velocity, then the equation for the head causing flow through the spray nozzles into the condenser will be found from equation (40):

$$H_1 = 1{\cdot}127(30 - p) - H - [(4fv^2)(l + 15nD)]/[2gD].$$

Example.—A parallel-flow jet condenser has to maintain a vacuum of 28 in. (barometer 30 in.). The cooling water is to be drawn from a pond having a temperature of 75° F., and the height from the inlet cooling water branch on the condenser to the water-level in the pond is 15 ft. The cooling water inlet pipe is 4 in. bore and 80 yd. long, and has four right-angled bends in it. The total area past the nozzles is 4 sq. in. Calculate the effective head, and find the quantity of cooling water which will be drawn into the condenser in gallons per minute, assuming a value for C_d of 0·75.

From equation (43) we have:

$$G = 2{\cdot}6 \times 4 \times 0{\cdot}75\sqrt{2gH_1}$$
$$= 62{\cdot}4\sqrt{H_1}.$$

Now v/v_v = nozzle area/pipe area,

or $v = v_v \times 4/12{\cdot}56;$

also $v_v = \sqrt{2gH_1},$

or $H_1 = v_v^2/2g.$

Then, substituting these values in equation (40), we get:

$$
\begin{aligned}
v_v^2/2g &= 1{\cdot}127(30-2) - 15 - [(4 \times 0{\cdot}01 \times v_v^2 \times 0{\cdot}319^2 \times 12)/(64{\cdot}4 \times 4)] \\
&\qquad \times [240 + (15 \times 4 \times 4)/12] \\
&= 31{\cdot}56 - 15 - 0{\cdot}0493v_v^2; \\
v_v^2 &= 256; \\
v_v &= 16 \text{ ft. per second.}
\end{aligned}
$$

Then, substituting these values in the equation for H_1, we get:

$$
\begin{aligned}
H_1 &= 256/64{\cdot}4 \\
&= 3{\cdot}97 \text{ ft.}
\end{aligned}
$$

The quantity of cooling water drawn in will be:

$$
\begin{aligned}
G &= 62{\cdot}4\sqrt{H_1} \\
&= 62{\cdot}4 \times \sqrt{3{\cdot}97} \\
&= 124{\cdot}3 \text{ gall. per minute.}
\end{aligned}
$$

Example.—A parallel-flow, low-level jet condenser is to maintain a vacuum of 29 in. with cooling water at 50° F. The quantity of steam to be condensed is 8000 lb. per hour, and the steam is 89 per cent dry. The low-water level in the river from which the water is to be drawn is 8 ft. below the water inlet branch on the condenser. The cooling water pipe has a 7-in. bore and is 40 ft. long, and there are six easy bends in it. Calculate the necessary area past the spray nozzles. Assume that the outlet water temperature is 3° F. lower than the vacuum temperature.

First, find the quantity of cooling water necessary to condense the steam. From the equation (37) we have:

$$
\begin{aligned}
G &= [W(Lq + t_s - T_2)]/[600(T_2 - T_1)], \\
\text{i.e.} \quad G &= [8000(1047{\cdot}3 \times 0{\cdot}89 + 79 - 76)]/[600(76 - 50)] \\
&= 479 \text{ gall. per minute.}
\end{aligned}
$$

The water velocity through the cooling water inlet pipe is,

$$
\begin{aligned}
v &= (4790 \times 144)/(62{\cdot}3 \times 60 \times 38{\cdot}5) \\
&= 4{\cdot}79 \text{ ft. per second.}
\end{aligned}
$$

From equation (40) we find:

$$
\begin{aligned}
H_1 &= 1{\cdot}127(30 - 1) - 8 - [\{4 \times 0{\cdot}01(4{\cdot}78)^2 \times 12\}/(2g \times 7)] \\
&\qquad \times [40 + (15 \times 6 \times 7)/12], \\
\text{i.e.} \quad H_1 &= 22{\cdot}43 \text{ ft.}
\end{aligned}
$$

From equation (43) we have:

$$
\begin{aligned}
G &= 2{\cdot}6NC_dA\sqrt{2gH_1}, \\
\text{or} \quad NA &= G/(2{\cdot}6C_d\sqrt{2gH_1}).
\end{aligned}
$$

Taking $C_d = 0{\cdot}9$ we get:

$$
\begin{aligned}
NA &= 479/(2{\cdot}6 \times 0{\cdot}9 \times 38) \\
&= 5{\cdot}39 \text{ sq. in.}
\end{aligned}
$$

AIR COOLERS IN PARALLEL-FLOW CONDENSERS

An air cooler is usually fitted to a parallel-flow condenser when the condenser is designed for a high vacuum. As will be explained later the cooler is not so necessary for low vacua. The cooler is usually formed by a portion of the steam space being partitioned off and fitting a water jet or spray to the top of this portion, the jet being discharged downwards while the air and vapour pass upwards, thereby giving a contra-flow effect to the water and air. The air suction branch is taken from above the jet where the temperature is least. The cooling water is usually taken from the main cooling water inlet supply. The problems of air cooling and vapour condensing follow the same laws and rules as those given for the contra-flow condenser on pp. 78 and 79.

HICK-BREGUET LOW-LEVEL JET CONDENSER

Fig. 38 shows a section of a Hick-Breguet low-level, parallel-flow jet condenser as manufactured by Messrs. Hick Hargreaves, Bolton, under Delaportes Patent. The cooling water enters by the branch A and is sprayed through the spray nozzles B by the vacuum; the steam enters by C and, mixing with the finely divided spray from the nozzles, is condensed and falls with the injection or cooling water to the large sump D formed in the bottom of the condenser. The water is withdrawn by the rotary water extraction pump E and discharged to the pond or river. The air is withdrawn through the cooler F, via the outlet branch G, by the Hick-Breguet ejector H. The water spray for the air cooler is shown at J, while K is the partition forming the cooler. L is a hollow float operating the water sealed vacuum breaker valve M. The air which breaks the vacuum in the condenser enters by the external pipe N, so that this air enters the condenser above the water-level quite independently of the height to which the water may rise.

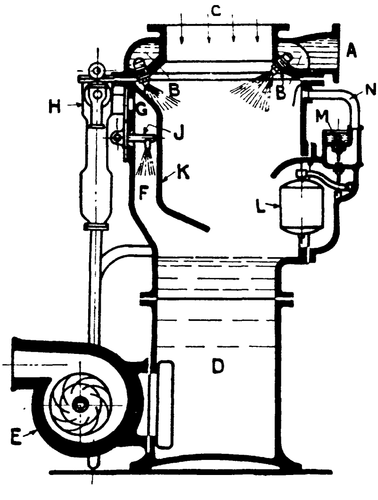


Fig. 38.—Hick-Breguet Low-level Jet Condenser

The special features and advantages claimed by the makers for this condenser are:

1. The brass nozzles spraying the injection or cooling water into the condenser in the form of two intersecting cones ensure complete mixture of the steam and water.

2. The injection or cooling water being admitted through annular chambers to the spray nozzles gives a clear passage for the exhaust steam.

3. The design of the vacuum breaker ensures the free access of air to the condenser quite independently of the height to which the water-level may rise.

Fig. 39 indicates how effectively these spray nozzles break up the cooling water to give a finely divided spray, so that as large a water surface as possible is presented to the steam; the nozzles are spiral and made of a special bronze, which is claimed to resist the eroding action of the water for many years.

These condensers are guaranteed by their makers to give a vacuum of 99·5 per cent of the theoretical vacuum on load, and the temperature of the condenser is guaranteed to be uniform throughout.

The following table gives the test figures of one of the above multi-jet type condensers manufactured by Messrs. Hick Hargreaves, Bolton. The condenser was working in conjunction with a 2000 kw. turbo-alternator.

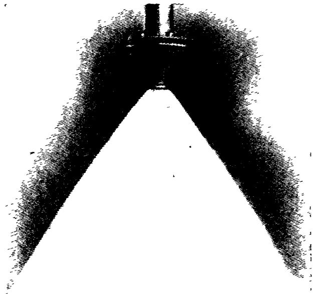


Fig. 39—Nozzle for Hick-Breguet Jet Condenser

	Load in Kilowatts.	Vacuum in Inches Hg.	Barometer in Inches Hg.	Temperatures.		Quantity of Cooling Water, Gallons per Min.
				Cooling Water Inlet, Degrees F.	Discharge, Degrees F.	
Figures guaranteed	2000	27·5	30	80	—	—
Figures attained on test	2000	27·8	30	83	100	1933

Taking the test figures, the vacuum corresponding to 100° F. is 28·05 in. Hg, and the attained vacuum was 27·8 in. Hg.

The percentage vacuum attained of the theoretical is therefore

$$= (27{\cdot}8 \times 100)/28{\cdot}05 = 99{\cdot}11 \text{ per cent.}$$

The partial air pressure is $28{\cdot}05 - 27{\cdot}8 = 0{\cdot}25$ in. Hg.

The coefficient of performance (see p. 47)

$$\begin{aligned} &= (T_2 - T_1)/(T_s - T_c + 10) \\ &= (100 - 83)/(104 - 100 + 10)\,. \\ &= 1{\cdot}214. \end{aligned}$$

MIRRLEES LOW-LEVEL JET CONDENSERS

Fig. 40 shows in section the latest type of low-level jet condenser as manufactured by Messrs. The Mirrlees Watson Company.

This condenser is of the parallel-flow type with integral water extraction pumps and air pump situated at the bottom. The exhaust steam enters at the top A, and immediately under the exhaust inlet flange an injection water inlet belt or chamber B is arranged round the exhaust pipe. S is the injection water inlet. A cylindrical skirt C is fitted inside the shell, and is perforated with holes D, so that the injection water is discharged into the body of the steam and ensures complete intermingling. The combined mass of injection water and steam falls through the cone piece E, where the greater part of the steam is condensed, to the bottom of the condenser and enters the impellers of the double water-extraction pumps F. The air, vapour, and non-condensable gases receive a compression in the cone E

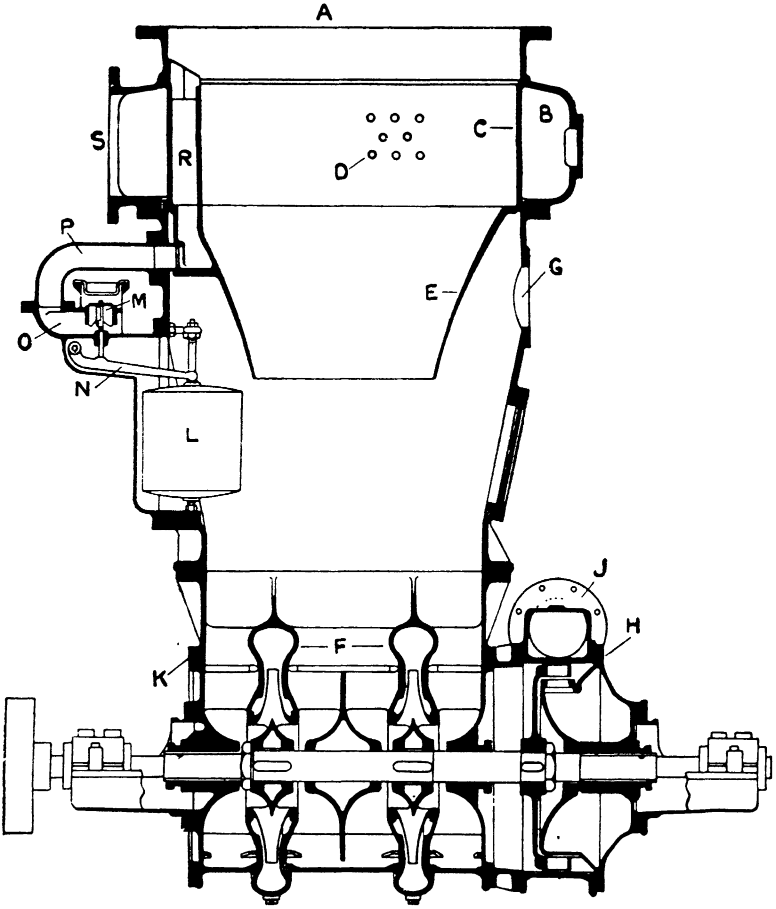


Fig. 40.—Mirrlees Low-level Jet Condenser

by the velocity of the incoming steam, and then flow upwards outside the cone piece E to the air suction G. The air pump H is of the Mirrlees-Le Blanc rotary water-jet type, and is arranged on the same shaft as the water-extraction pumps and driven by the same motor. The air pump suction branch J is connected by cast-iron piping to the condenser air suction G, and no integral air cooler is fitted. With this type of air pump the necessity for such a cooler disappears, because the air pump is of the water-jet type, and being such its supply of jet water is cold. This cold water performs the function of an air cooler by condensing a large proportion of the associated vapour, and at the same time cooling the air. This arrangement of integral water-extraction pumps gives a compact arrangement by avoiding the necessity for external piping between the bottom of the condenser and the water-extraction pumps; it also avoids the common trouble of the extraction pumps becoming deprimed by air leakage at the

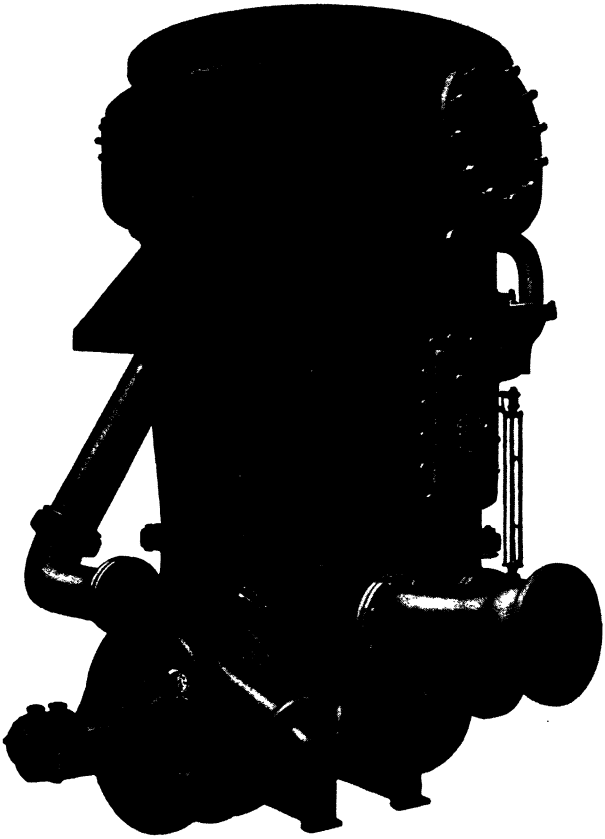


Fig. 41.—Mirrlees Multi-jet Condenser, equipped with Mirrlees-Leblanc Rotary Air Pump and Centrifugal Water-extraction Pump with Coupling for Connection to Electric Motor, Steam Engine, or Steam Turbine.

glands. Should such leakage take place with this arrangement, the air leaks into the condenser direct, and will thus pass to the air suction instead of having to flow along the suction pipe to the condenser in a direction opposite to the flow of the water, thus damming the water back in the condenser and preventing its free flow to the extraction pump. A large end cover K is fitted, so that the impellers are easily withdrawn for overhaul. A float operated automatic vacuum breaker is fitted to act if the water rises to a dangerous height. L is a hollow copper float which will rise on the surface of the rising water, and open the valve M by means of the lever N and admit atmospheric air to the chamber O. The chamber O is in communication with the top of the condenser by the pipe P and the duct R, formed in the skirt C and cone piece E. When the float rises atmospheric air will be drawn into the top of the condenser and above the surface of the water, so that the air meets with no undue resistance; the vacuum therefore will be quickly broken.

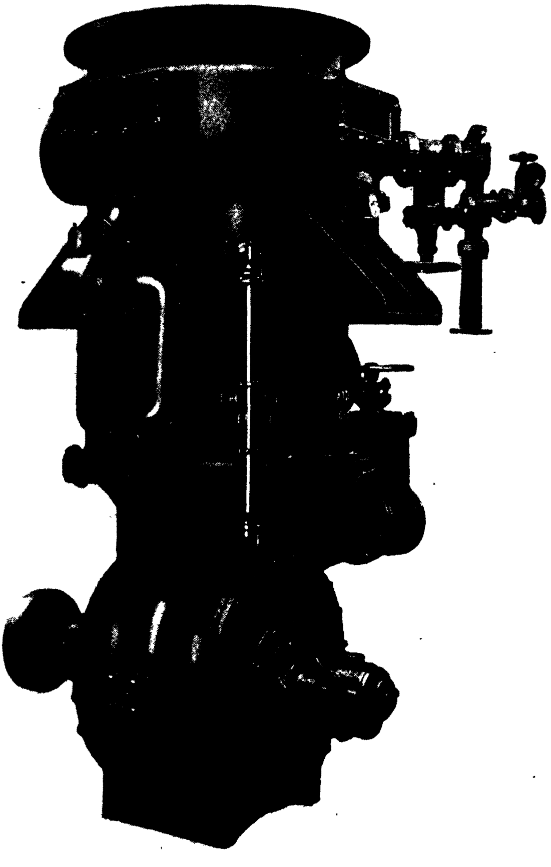


Fig. 42.—Mirrlees Multi-jet Condenser

Fig. 41 (p. 93) shows a photograph of one of these low-level jet condensers, taken from the air pump end.

When the conditions of the station warrant, Messrs. The Mirrlees Watson Company fit their steam-jet operated air ejectors to these low-level jet condensers in place of the rotary water-jet air pumps. In this case, however, they arrange a small air cooler or devaporizer integral with the condenser and in a manner similar to that described on p. 90.

Fig. 42 shows the arrangement when a steam-jet operated air ejector is fitted. This is a smaller plant than that shown in fig. 41, and only a single impeller is employed on the water-extraction pump.

These condensers are designed for a temperature difference of 4° F. (see p. 78), between the temperature of the discharge water and the temperature corresponding to the vacuum, when the difference between the inlet water temperature and the vacuum temperature does not exceed 40° F. For differences of from 40° F. to 50° F. the temperature difference between the discharge water and the vacuum temperature is 5° F., and from 50° F. to 60° F. the difference is 7° F., and from 60° F. to higher temperatures the difference is 9° F. The heat absorbed by the cooling water is assumed to be 1000 B.Th.U. per pound of steam condensed.

The following figures taken at site from a large Mirrlees-Le Blanc multijet condensing plant have been supplied to the author by Messrs. The Mirrlees Watson Company.

Barometer in Inches Hg.	Vacuum in Inches Hg.	Vacuum Corrected to 30 in. Hg Barometer, Inches Hg.	Temperature Corresponding to Corrected Vacuum, Degrees F.	Inlet Water Temperature, Degrees F.	Outlet Water Temperature, Degrees F	Load in Kilowatts.
29·5	28·5	29·0	79·3	48	70	3000
29·5	28·4	28·9	82·5	49	76	3850
29·5	28·3	28·8	85·0	49	81	4500
29·49	28·2	28·71	87·0	49	84	5350

The full load condition for the plant was 6000 kw. The vacuum producing apparatus was a Mirrlees steam-ejector air pump.

The partial air pressure at the highest load is:

$$\begin{aligned} &\text{vacuum corresponding to } 84^\circ \text{ F.} - 28{\cdot}71 \text{ in. Hg,} \\ &= 28{\cdot}83 - 28{\cdot}71 \\ &= 0{\cdot}12 \text{ in. Hg.} \end{aligned}$$

This low partial air pressure shows the ability of the air pump to reduce the air pressure to the lowest limits, and indicates that the plant is air-tight.

The depression of the condensate temperature below the vacuum temperature is at the highest load 87 − 84 = 3° F., and at the lowest load 79·3 − 70 = 9·3° F. This difference of condensate depression temperature indicates the presence of a greater quantity of air entering the condenser at the lower loads; this is also proved by the air pressure at the lowest load which is,

$$\begin{aligned}&\text{vacuum corresponding to } 70^\circ \text{ F.} - 29 \text{ in. Hg,}\\ &= 29{\cdot}26 - 29\\ &= 0{\cdot}26 \text{ in. Hg.}\end{aligned}$$

The diminution of the partial air pressure as the load increases is normally found to take place, and is caused by the more efficient steam sealing of the turbine glands at the higher loads, and by the point along the expansion curve at which the pressure is atmospheric travelling towards the L.P. end of the turbine as the load rises, thus placing fewer joints under vacuum.

The coefficient of performance of this condenser is at the highest load

$$\begin{aligned}&= (T_2 - T_1)/(T_s - T_c + 10)\\ &= (84 - 49)/(87 - 84 + 10)\\ &= 2{\cdot}69.\end{aligned}$$

EXAMPLES ON JET CONDENSERS

1. A low-level contra-flow condenser is supplied with cooling water at a temperature of 60° F., the difference between the temperature of the water being discharged by the extraction pump and the vacuum temperature is 3° F. The recorded vacuum is 28·1 in. Hg, and the barometer reading is 29·65 in. Hg. The heat contents of the steam entering the condenser as measured by a calorimeter were 1010 B.Th.U. per pound reckoned from freezing-point. Find the ratio between the quantity of cooling water entering the condenser for condensing purposes and the quantity of steam condensed.

Answer.—31·73.

2. If in the above example the total quantity of air being dealt with by the air pump is 80 lb. per hour; the quantity of steam being condensed 75,000 lb. per hour; and the temperature of the air and vapour leaving the condenser is 70° F.; find the ratio between the total quantity of cooling water entering the condenser and the quantity of steam condensed.

Answer.—31·96.

3. In a low-level multi-jet condensing plant the water-level in the pond is 6 ft. above the water inlet branch on the condenser. The inlet water pipe is 7 in. bore, 40 ft. long, and there are six easy right-angled bends in it. The vacuum in the condenser is 28 in. Find the pressure drop across the spray nozzles or nozzle head when the quantity of water flowing into the condenser is 500 gall. per minute.

Answer.—34·964 ft.

4. Find the total area of spray nozzles necessary in the condenser given in question 3, assuming a coefficient of discharge of 0·85.

Answer.—4·76 sq. in.

5. In a low-level multi-jet condensing plant, having a separate air-cooling compartment, the temperature of the inlet cooling water is 60° F. The vacuum in the condenser is 28 in. Hg, and the water leaving the condenser is 2° F. lower than the vacuum temperature. The air suction temperature is 70° F. Find the ratio between the quantity of cooling water entering the cooler and the mass of air which is being dealt with by the air pump. The barometer reading is 29·813 in. Hg.

Answer.—286·6.

CHAPTER III

High-level Jet Condensers

The high-level or the barometric condenser is so named because it is placed at a height greater than that of the water barometer. The water, therefore, will flow out of the condenser by gravity and against the vacuum in the condenser. There is thus no need for a water-extraction pump, but a pump must be fitted to discharge the cooling water up to the water inlet branch on the condenser.

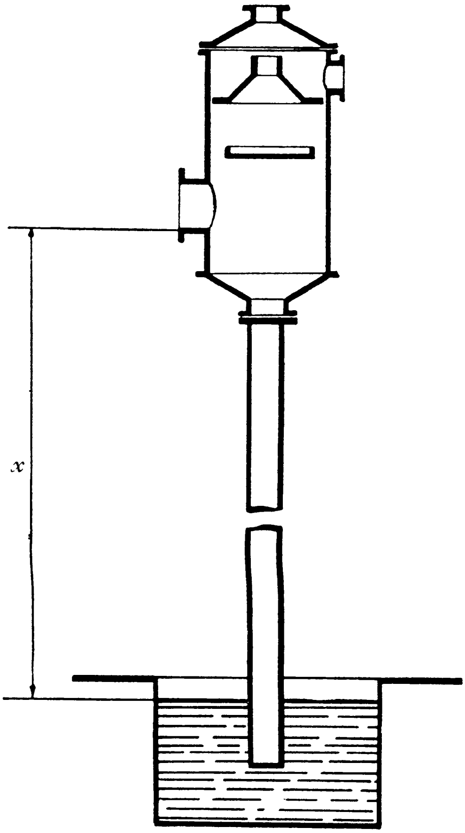


Fig. 43.—High-level Jet Condenser

The condenser head as it is called, or the condenser proper, is supported on a high staging, and in general design is similar to either the low-level contra-flow or parallel-flow types. Some makers use the same head for their high- and low-level plants, and for the parallel-flow type add a cone piece to the bottom. The water outlet or tail pipe extends from the bottom of the condenser to the ground-level, and has its lower end immersed in a sump.

The rules already given for low-level plants for the necessary quantity of cooling water, the volume of the shell, the air cooler, &c., and all particulars pertaining to the condenser head itself are applicable to this type. The main new problems which this type evokes are the height at which the condenser head must be placed for efficient drainage and the capacity of the sump into which the tail pipe discharges.

The capacity of the sump should be such that, when a pump is fitted to draw the water from the sump and discharge it to a cooling tower or pond, the pump should have at least one minute's supply before drawing air; the depth to which the tail pipe is immersed should be such that, on starting up, the water-level over the end of the tail pipe should not be less than 1 ft. 6 in. after the vacuum has drawn the water up and filled the tail pipe.

The height at which the condenser head must be placed to ensure that the water will not be drawn up by the vacuum and get down the exhaust pipe can be found as follows:

Let fig. 43 represent a condenser.

Let x be the height in feet from the bottom of the exhaust pipe to the water-level in the sump;
p, the absolute pressure in the condenser in inches of mercury;
F, the friction loss in the tail pipe in feet;
D, the diameter of the tail pipe in feet; and
v, the water velocity in the tail pipe in feet per second;
s, the safety height to ensure water not getting down the exhaust pipe. (Usually taken at 1 ft. 6 in. to 2 ft.) This height also includes an allowance for the decrease in density of the water owing to increase in temperature.

Since the water flows against the vacuum pull in the condenser, all the losses in friction, in exit velocity, &c., must be added to the vacuum pull, so that reckoning on a standard barometer of 30 in. Hg we get:

$$x = 1{\cdot}127(30 - p) + \mathrm{F} + s + v^2/(2g).$$

As before, and making the length of the tail pipe $l = x$, we get:

$$\mathrm{F} = \{(4fv^2)/(2g\mathrm{D})\}(x + 15n\mathrm{D}),$$

n being the number of bends in the tail pipe.

Then, substituting in the above equation, we get:

$$x = 1{\cdot}127(30 - p) + s + \{v^2/(2g)\}[1 + (4f/\mathrm{D})(x + 15n\mathrm{D})]\ldots.(44)$$

Some makers reckon a full barometric head of 34 ft. of water in place of the actual condenser vacuum, so that on this reckoning the above formula would be:

$$x = 34 + s + \{v^2/(2g)\}\{1 + (4f/\mathrm{D})(x + 15n\mathrm{D})\},$$

which on reduction becomes

$$x = [34 + s + \{v^2/(2g)\}(1 + 60fn)]/[1 - (4fv^2)/(2g\mathrm{D})]\ldots..(45)$$

Example 1.—A barometric condenser is to maintain a vacuum of 25 in. Hg, when dealing with 8000 lb. of exhaust steam per hour, having a dryness of 85 per cent. The condensate temperature is to be maintained 4° F. lower than the vacuum temperature, with cooling water at 70° F. Allowing a water velocity in the tail pipe of 5 ft. per second, and a safety height of 2 ft., calculate the height x at which the condenser head must be placed. The tail pipe has two easy right-angled bends in it.

The quantity of cooling water will be found from equation (37) (p. 78)

$$\begin{aligned} G &= [W(qL + t_1 - T_2)]/[600(T_2 - T_1)] \\ &= [8000(1021{\cdot}1 \times 0{\cdot}85 + 126 - 122)]/[600(122 - 70)] \\ &= (8000 \times 860)/(600 \times 52) \\ &= 223{\cdot}5 \text{ gall. per minute.} \end{aligned}$$

The diameter of the tail pipe will be:

$$\begin{aligned} (\pi/4) \times D^2 &= \frac{\text{volume of water passing in cubic feet per second}}{\text{velocity of water in feet per second}}, \\ \text{i.e. } D &= \sqrt{[4(2235/60 + 8000/3600)]/[62{\cdot}3 \times 5 \times \pi]} \\ &= \sqrt{0{\cdot}1614} \\ &= 0{\cdot}4 \text{ ft.} \\ &= 4{\cdot}8 \text{ in.} \end{aligned}$$

Then, putting $s = 2$ ft., we get from equation (44):

$$\begin{aligned} x &= 1{\cdot}127(30-4)+2+\{5^2/(2g)\}[1+(4\times 0{\cdot}01/0{\cdot}4)(x+15\times 2\times 0{\cdot}4)] \\ &= 29{\cdot}3 + 2 + 0{\cdot}854 + 0{\cdot}0388x, \\ \text{i.e. } x &= 33{\cdot}5 \text{ ft.} \end{aligned}$$

Example 2.—A barometric jet condenser has a water-level gauge-glass fitted to the shell. The water-level shown in the glass measures 34 ft. from the water-level in the discharge sump when the engine, exhausting to the condenser, is developing 1200 i.h.p. The bore of the tail pipe is 5 in., and the vacuum 25·5 in. Hg. The barometer reading is 29·5 in. Hg. The temperature of the cooling water entering the condenser is 60° F., and the temperature of the water in the tail pipe 122° F. Deduce from the above particulars the quantities of cooling water and exhaust steam entering the condenser, assuming that the dryness fraction of the exhaust is 0·85. The tail pipe has no bends in it.

The height x will equal the vacuum head plus the velocity head plus the friction head,

$$\text{i.e.} \qquad x = 1{\cdot}127(30 - p) + \{v^2/(2g)\}(1 + 4fx/D).$$

The absolute pressure p in the condenser will be $29{\cdot}5 - 25{\cdot}5 = 4$ in. of mercury.

By substituting we get:

$$\begin{aligned} 34 &= 1{\cdot}127(30 - 4) + \{v^2/(2g)\}[1 + (4 \times 0{\cdot}01 \times 34 \times 12)/5] \\ &= 29{\cdot}3 + 0{\cdot}0662v^2, \\ \text{or} \qquad v &= \sqrt{4{\cdot}7/0{\cdot}0662} \\ &= \sqrt{71} \\ &= 8{\cdot}43 \text{ ft. per second.} \end{aligned}$$

Since the tail pipe has a 5-in. bore, the total quantity of water Q passing down the tail pipe will be

$$\begin{aligned} Q &= (19{\cdot}63 \times 8{\cdot}43 \times 6{\cdot}23 \times {\cdot}60)/144 \text{ gall. per minute} \\ &= 430 \text{ gall. per minute.} \end{aligned}$$

From equation (37),

$$\begin{aligned} G &= [W(qL + t_1 - T_2)]/[600(T_2 - T_1)] \\ &= [W(0{\cdot}85 \times 1021{\cdot}1 + 126 - 122)]/[600(122 - 60)]. \\ &= (870W)/(600 \times 62) \\ &= 0{\cdot}0233W. \\ \text{Now } Q &= G + W/600, \\ \therefore G &= 430 - W/600. \end{aligned}$$

Then equating these two values of G we get:

$$\begin{aligned} 0{\cdot}0233W &= 430 - W/600, \\ \text{i.e.} \qquad W &= 18{,}428 \text{ lb. per hour;} \end{aligned}$$

and G will be:

$$\begin{aligned} G &= 430 - 18{,}428/600 \\ &= 399 \text{ gall. per minute.} \end{aligned}$$

HORSE-POWER ABSORBED BY CONDENSING WATER PUMP FOR BAROMETRIC JET CONDENSER

Let G be the quantity of condensing water in gallons per minute;
p, the absolute pressure in the condenser in inches of mercury;
H, the height of water inlet branch on condenser to water-level in river or pond in feet;
l, the total length of suction and delivery piping;
D, the internal diameter of piping in feet;
F, the friction loss;
v, the water velocity in feet per second; and
H_E, the effective head against which the pump discharges in feet.

There are two cases to be considered here, one when the vacuum has been created in the condenser, and the other when there is no vacuum. The first is the normal condition.

Case 1.—The effective head H_E will be:

$$H_E = H - 1{\cdot}127(30 - p) + F + \text{loss at exit.}$$

As before $F = \{(4fv^2)/(2gD)\}(l + 15nD)$,

and the loss at exit $= v^2/2g$.

$$\therefore H_E = H - 1{\cdot}127(30 - p) + \{(4fv^2)/(2gD)\}\{l + 15nD\} + v^2/(2g).$$

The water horse-power will be:

$$\begin{aligned} \text{w.h.p.} &= (10GH_E)/33{,}000, \\ &= [10G\{H - 1{\cdot}127(30 - p) + \{(4fv^2)/(2gD)\}(l + 15nD) + v^2/(2g)\}] \\ &\quad \div 33{,}000. \end{aligned}$$

If the pump efficiency be E the b.h.p. at the pump coupling will be:

$$\text{b.h.p.} = [10G\{H - 1{\cdot}129(30 - p) + (4fv^2/2gD)(l + 15nD) + v^2/(2g)\}] \div (33{,}000E).$$

Example 3.—Assuming in the previous example that the height H was 40 ft., the pump efficiency 70 per cent, and that the pump suction and discharge piping was 5 in. bore and 50 ft. long with four bends in it: find what the b.h.p. absorbed at the pump coupling would be.

The velocity through the 5-in. bore piping will be:

Case 1

$$\begin{aligned} v &= 8{\cdot}4 \times 399/430 \\ &= 7{\cdot}78 \text{ ft. per second.} \end{aligned}$$

Then the b.h.p. absorbed will be:

$$\begin{aligned} \text{b.h.p.} &= [(10 \times 399)/(33{,}000 \times 0{\cdot}7)] \\ &\quad \times [40 - 1{\cdot}127 \times 26 + \{4 \times 0{\cdot}01 \times 60{\cdot}6 \times 12(50 + \tfrac{1}{12} \times 15 \times 4 \times 5)/(64{\cdot}4 \times 5)\} + 60{\cdot}6/64{\cdot}4] \\ &= (10 \times 399 \times 18{\cdot}382)/(33{,}000 \times 0{\cdot}7) \\ &= 3{\cdot}16. \end{aligned}$$

Case 2 When no vacuum is in the condenser we get:

$$\begin{aligned} \text{b.h.p.} &= [(10 \times 399)/(33{,}000 \times 0{\cdot}7)] \\ &\quad \times [40 - 0 + \{4 \times 0{\cdot}01 \times 60{\cdot}6 \times 12(50 + \tfrac{1}{12} \times 15 \times 4 \times 5)/(64{\cdot}4 \times 5)\} + 60{\cdot}6/64{\cdot}4] \\ &= (10 \times 399 \times 47{\cdot}732)/(33{,}000 \times 0{\cdot}7) \\ &= 8{\cdot}2. \end{aligned}$$

Case 1 is the normal condition, but to allow for a drop in vacuum, due to any cause whatever, the motor driving the pump should be capable of withstanding a considerable overload without undue heating of the coils.

The above example shows that if the vacuum be entirely lost, and if the pump characteristics are such that the normal quantity is maintained against the increased head, the motor will be $(8{\cdot}2 \times 100)/3{\cdot}16 = 260$ per cent overload. The characteristics of the pump, however, would be such that there would be a reduction in the quantity, so that the maximum b.h.p. would be less than that given above, the exact power depending upon quantity head characteristics of the pump.

MIRRLEES HIGH-LEVEL JET CONDENSER

Fig. 44 shows the latest design of high-level jet condenser as manufactured by Messrs. The Mirrlees Watson Company.

The condenser is of the parallel-flow type.

The exhaust steam enters at A, and the injection water at B. The injection water flows round the injection chamber C, and, passing through the holes in the perforated skirt D, is discharged into the body of the exhaust steam entering at A, thus ensuring complete intermingling of steam and water. The combined mass of injection water and steam passes down through the condenser cone E where the greater part of the steam is condensed, and the air and non-condensable gases are compressed by the velocity of the incoming steam. The condensed steam and injection water leave by the outlet F, to which is bolted the long tail pipe. The air and vapour flow upwards round the outside of the condenser cone E, and leave by the outlet G. To this outlet G is bolted a separator H which separates the heavy particles of moisture carried over with the air.

The mixture of air and vapour is given a sudden reversal of flow by the internal pipe J, the heavy moisture passing downwards to the drain L which is connected to the tail pipe. The air and its associated vapours pass upwards round the internal pipe J and leave by the air suction K, which is connected to the air pump. On the condenser bottom cover M there are cast ribs N to destroy the free vortex, which would be set up were these ribs not there. This enables the condenser to be set at the correct height for a still water-level, and ensures that the water does not rise up the side of the condenser shell and into the outlet G. The condenser and separator are made of close-grained cast iron throughout.

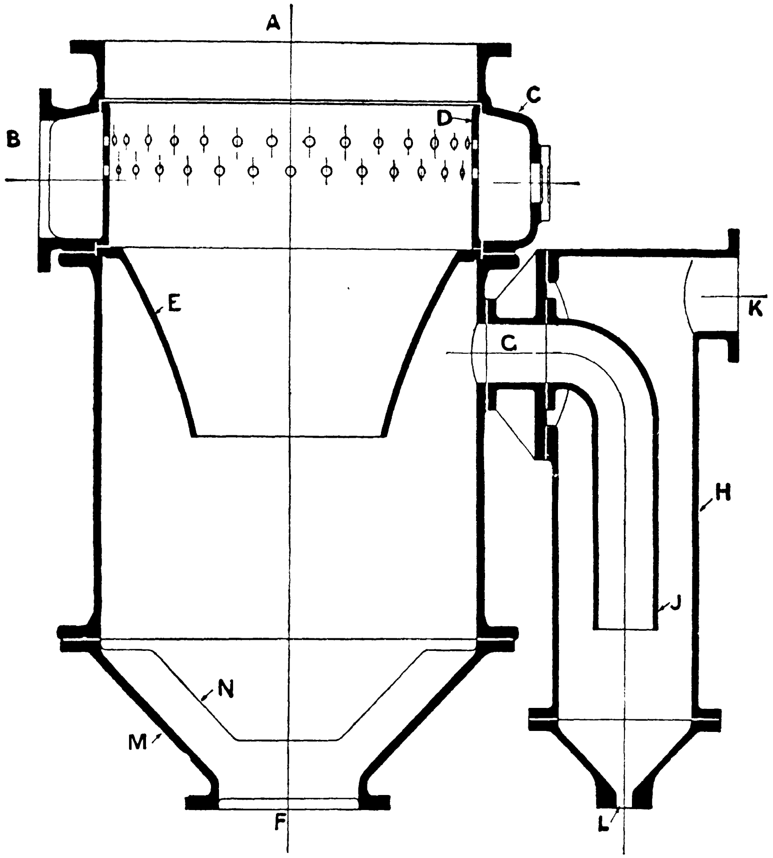


Fig. 44.—Diagrammatic Sectional Arrangement of Parallel-flow High-level Jet Condenser

CHAPTER IV

Ejector Condensers

The ejector condenser consists essentially of a water jet discharging through a series of guide cones, so arranged to guide the steam on to the surface of the water jet; the water jet enters a diffuser nozzle, where the velocity energy is converted into pressure energy and the water is discharged against the vacuum pull. These condensers require no independent air pump, because of the air entraining effect of the water jet itself.

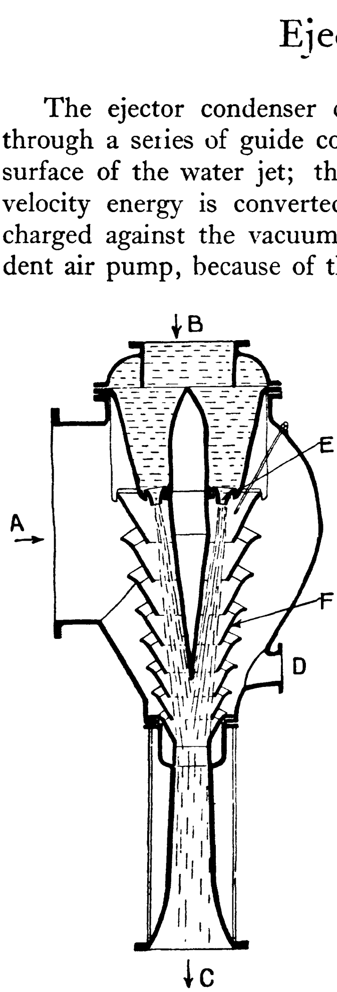


Fig. 45.—Körting Multi-jet Ejector Condenser

Fig. 45 shows the latest design of a multi-jet ejector condenser manufactured by Messrs. Körting Brothers. A is the exhaust steam inlet, B the condensing water inlet, and C the discharge outlet. D is a connection to which a vacuum breaker is fitted.

The condensing jet consists of a number of nozzles E, arranged concentrically and having their axis inclined at such an angle that the water jet assumes the form of an inverted cone. Surrounding the water jet are the guide cones F, arranged with an increasing area from the bottom to the top. In the upper part of the condensing cone the water will be colder than it is lower down, so that the temperature difference between the steam and the water at the top will be greater than it is lower down, and hence the condensation will be greatest at the top and gradually diminish to zero at the bottom. The guide cones are therefore arranged with an increasing area from the bottom upwards, the proportions being such that the steam velocity through the cones is constant from top to bottom. It will be realized that the vacuum inside the guide cones F will be higher than it is in the exhaust chamber, and with the above arrangement of guide cones giving a uniform or constant steam velocity the vacuum drop will also be uniform and can thus be reduced to a minimum. The magnitude of this vacuum drop varies with the actual vacuum; at 28 in. Hg in the exhaust chamber a drop of 0·5 in. Hg is sufficient to impart the necessary velocity to the steam. When the steam strikes the jet it is immediately condensed and the particles of water, into which it is converted, having the kinetic energy of their equivalent weight of steam, cut into the jet and contribute to the momentum of the jet, thus assisting in the discharge against the atmosphere. The air which enters the condensing cone along with the steam has the steam velocity and thus combines easily with the water. The air is discharged along with the water.

In contradistinction to the spray nozzles used in parallel-flow jet condensers where the nozzles are designed to break up the water into a fine spray, the nozzles in these multi-jet ejector condensers are designed to give a solid smooth jet of water, so that the eddy losses are reduced to a minimum, and thereby reduce the work of compression in the discharge diffuser to a minimum. This smooth solid jet has the capacity of retaining the air and other gases which are in solution in the condensing water, so that the air which the jet has to entrain is only that due to leakage and that which comes over with the steam. Spray nozzles always liberate the air in solution, and this liberated air has to be dealt with by air pumps.

To ensure satisfactory working under all conditions of load, the ejector condenser must be supplied with condensing water at a pressure at the inlet branch on the condenser of not less than 20 ft. head of water. This head is usually derived from a centrifugal circulating pump, which may be steam, motor, or belt driven, and which draws the water from a river or pond and discharges direct to the condenser. When direct-acting circulating pumps are employed they should discharge to an overhead tank placed at a height of not less than 20 ft. plus the friction loss in the pipe, above the condenser inlet; from the overhead tank the water flows direct to the condenser. This arrangement gives a steady flow to the condenser, and is preferable to the direct-acting pumps discharging direct to the condenser.

The ratio between the weight of condensing water necessary to give a specific vacuum to the weight of steam condensed is practically the same as that necessary for surface or jet condensers working under the same conditions of vacuum and water temperature.

A 1000-kw. reciprocating steam-engine and generator, having a steam consumption of 20,000 lb. steam per hour, would require for a multi-jet Körting ejector condenser 55,000 gall. of condensing water per hour at 60° F. to maintain a vacuum of 26 in. Hg. This gives a ratio of water to steam of 27·5. The condenser would be 6 ft. long, and allowing a pressure at the water inlet branch of 20 ft. head of water, the head against the circulating pump would be 6 + 20 + friction losses in the pipe line + the difference of level between the pump intake and the condenser outlet flange. Assuming an allowance of 6 ft. to be sufficient for these last two items the total lift would be 32 ft.; allowing a combined efficiency of 50 per cent for a motor-driven centrifugal circulating pump the h.p. necessary for supplying the condenser with condensing water is:

$$\begin{aligned} \text{h.p.} &= (550{,}000 \times 32)/(60 \times 33{,}000 \times 0{\cdot}6) \\ &= 14{\cdot}82, \text{ or } 11{\cdot}05 \text{ kw.} \end{aligned}$$

This as a percentage of the full load output of the set is

$$\begin{aligned} &(11{\cdot}05 \times 100)/1000 \\ &= 1{\cdot}105 \text{ per cent.} \end{aligned}$$

With a turbine of the same size but requiring a vacuum of 28 in. Hg a condenser using 88,000 gall. of water per hour would be required. The percentage of power required for this set would be

$$\begin{aligned} &(1{\cdot}105 \times 88)/55 \\ &= 1{\cdot}77 \text{ per cent.} \end{aligned}$$

With these condensers it is inadvisable to use cooling water at a temperature above 75° F., as above this temperature the power required to drive the circulating pump becomes disproportionately high because of the large quantity of circulating water necessary. When the same water has to be used over again this can be accomplished either by spray cooling or by cooling towers, the former method being recommended by the makers of these condensers. They also recommend separate condensers for each steam unit, although the condensers work quite well when two or more engines are exhausting to a central condenser or when two condensers are working in parallel.

Many cases arise in practice where the ejector condenser is eminently suitable. Their adoption keeps down capital outlay, and, as they have no moving parts (if we neglect the circulating pump), wear and tear is practically *nil* even when dirty river or canal water is used. They occupy small space, are certain in action, and the cost of working is low.

To find the velocity of efflux from the water nozzles:

Let H be the water head at the inlet in feet of water;
p, the absolute pressure inside the guide cones, in inches of mercury;
v, the velocity of efflux;
C_v, the coefficient of velocity = say 0·95; and
H_E, the effective head in feet in causing flow.

$$\begin{aligned} \text{Then } H_E &= 1{\cdot}127(30 - p) + H, \\ v &= C_v\sqrt{2gH_E} \quad \ldots\ldots (46) \\ &= C_v\sqrt{2g[1{\cdot}127(30 - p) + H]} \text{ ft. per second.} \quad \ldots\ldots (47) \end{aligned}$$

To find the velocity v_e at the inlet to the diffuser:

If M be the mass of steam condensed per second;
m, the mass of water passing per second;
v_s, the steam velocity in feet per second at the inside edge of guide cones (assumed constant from top to bottom);

v_e, the velocity in feet per second of the combined mass of water and condensed steam entering the diffuser; and

θ, the angle between the guide cones and the water jet, then we have:

the momentum of the jet entering the condenser $= mv$;

the momentum leaving the condenser and entering the diffuser

$$= (\mathrm{M} + m)v_e,$$

and the momentum added to the jet in its passage through the condenser is the sum of the momentum of the steam, plus the increased momentum of the jet, due to the work done by gravity on the water jet during its passage or fall through the condenser, or the added momentum

$$= \mathrm{M}v_s \cos\theta + m(v_e - v)$$
$$= \mathrm{M}v_s \cos\theta + m(\sqrt{v^2 + 2g\mathrm{H}} - v),$$

where H is the free height in the condenser through which the water falls.

The momentum leaving the condenser will equal the sum of the other momenta, i.e.

$$(\mathrm{M} + m)v_e = mv + \mathrm{M}v_s \cos\theta + m(\sqrt{v^2 + 2g\mathrm{H}} - v),$$
$$\text{or } v_e = \mathrm{K}[mv + \mathrm{M}v_s \cos\theta + m(\sqrt{v^2 + 2g\mathrm{H}} - v)]/(\mathrm{M} + m), \quad \ldots.(48)$$

where K is a coefficient to allow for losses due to eddies, &c.

The following diagram, fig. 46, will illustrate the various symbols.

Since ϕ is small its cosine may be taken as 1, and therefore v_e may be taken to be the velocity of the jet entering the diffuser parallel to its axis.

The velocity v_e is converted into pressure energy in the diffuser, so that the water is discharged against atmospheric pressure.

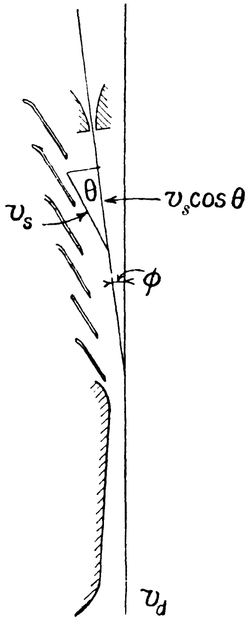


Fig. 46

RELATION BETWEEN THE VELOCITY v_e AND THE PRESSURE RISE IN THE DIFFUSER

Let v_d be the velocity at the discharge in feet per second;

p, the absolute pressure in inches of mercury at the inlet (assumed equal to the pressure inside the guide cones);

P, the absolute pressure in inches of mercury at the discharge; and

l_d, the length of the diffuser in feet.

Then, by Bernouilli's theorem, we have

$$1{\cdot}127p + v_e^2/(2g) = 1{\cdot}127\mathrm{P} + \{v_d^2/(2g)\} - l_d,$$
$$\text{i.e. } \{v_e^2/(2g)\} - v_d^2/(2g) = 1{\cdot}127(\mathrm{P} - p) - l_d. \quad \ldots\ldots\ldots(49)$$

If the efficiency of the diffuser is 80 per cent, this expression becomes

$$\{0{\cdot}8v_e^2/(2g)\} - v_d^2/(2g) = 1{\cdot}127(P - 0{\cdot}8p) - l_d.$$

$$\therefore\ v_d^2 = 2g[\{0{\cdot}8v_e^2/(2g)\} - 1{\cdot}127(P - 0{\cdot}8p) + l_d]. \ldots\ldots (50)$$

Example.—A multi-jet ejector condenser is to maintain a vacuum of 28 in. Hg in the exhaust pipe when supplied with 1500 gall. of condensing water at 60° F. The water pressure at inlet is 9 lb. per square inch gauge. Calculate the total area of the nozzles necessary, the velocity v_e, the area at the diffuser entrance or throat, the velocity of discharge v_d, and the area at the discharge. The angle θ is 30°, and the free height H in the condenser is 8 ft.

From equation (46) (p. 106):

$$\begin{aligned} v &= C_v\sqrt{2gH_E} \\ &= 0{\cdot}95\sqrt{64{\cdot}4[\{34 \times 9/14{\cdot}7\} + 1{\cdot}127(30 - 1{\cdot}5)]} \\ &= 55{\cdot}38 \text{ ft. per second.} \end{aligned}$$

The total area A_n of the nozzles will be:

$$\begin{aligned} A_n &= \frac{\text{quantity passing in cubic feet per second} \times 144}{\text{velocity in feet per second}} \\ &= (15{,}000 \times 144)/(62{\cdot}3 \times 60 \times 55{\cdot}38) \\ &= 10{\cdot}43 \text{ sq. in.} \end{aligned}$$

The steam velocity v_s will be that due to the heat drop across the guide cones, and, taking the vacuum drop to be 0·5 in. Hg, we get the heat drop from 28 in. to 28·5 in. Hg to be 4 B.Th.U (assuming a dryness fraction of 0·9). (See the Mollier or Hϕ diagram.)

$$\text{Then} \qquad v_s = \sqrt{Jh2g}, \ldots\ldots\ldots\ldots (51)$$

where J is Joule's mechanical equivalent of heat, and h the heat drop in B.Th.U. per pound, i.e.

$$\begin{aligned} v_s &= \sqrt{64{\cdot}4 \times 777{\cdot}8 \times 4} \\ &= 447{\cdot}6 \text{ ft. per second.} \end{aligned}$$

To find the total quantity of steam condensed we must first find the change in the dryness fraction of the steam passing through the guide cones. From the Mollier or Hϕ diagram this is 0·005 or the dryness fraction inside the guide cones is 0·895.

Then, from equation (37) (p. 78) we get:

$$\begin{aligned} G &= [W(qL + t_1 - T_2)]/[600(T_2 - T_1)], \\ \text{or } 1500 &= [W(0{\cdot}895 \times 1040 + 92 - 91)]/[600(91 - 60)], \\ \text{i.e. } W &= 29{,}943 \text{ lb. per hour.} \end{aligned}$$

The mass M of steam flowing per second:

$$\begin{aligned} M &= 29{,}943/3600 \\ &= 8{\cdot}3 \text{ lb. per second.} \end{aligned}$$

The mass m of water passing per second:

$$\begin{aligned} m &= 15{,}000/60 \\ &= 250 \text{ lb. per second.} \end{aligned}$$

Then from equation (48)

$$\begin{aligned} v_e &= \mathrm{K}[250\times 55{\cdot}38+8{\cdot}3\times 447{\cdot}6\times 0{\cdot}866+250(\sqrt{55{\cdot}38^2+64{\cdot}4\times 8}-55{\cdot}38)] \\ &\quad \div(8{\cdot}3+250) \\ &= 70{\cdot}34\mathrm{K} \text{ ft. per second.} \end{aligned}$$

Allowing a value for K = 0·75,

$$\begin{aligned} \text{then} \qquad v_e &= 0{\cdot}75 \times 70{\cdot}34 \\ &= 52{\cdot}75 \text{ ft. per second.} \end{aligned}$$

From equation (50) and assuming l_d to be 3 ft.,

$$\begin{aligned} v_d^2 &= 64{\cdot}4[\{(0{\cdot}8 \times 52{\cdot}75^2)/64{\cdot}4\} - 1{\cdot}127(30 - 0{\cdot}8 \times 1{\cdot}5) + 3] \\ &= 328{\cdot}44 \\ \therefore\ v_d &= 18{\cdot}12 \text{ ft. per second.} \end{aligned}$$

The total quantity of water flowing through the diffuser

$$\begin{aligned} &= \mathrm{M} + m \\ &= 8{\cdot}3 + 250 \\ &= 258{\cdot}3 \text{ lb. per second;} \end{aligned}$$

and the area A_{de} at the diffuser entrance or throat will be

$$\begin{aligned} \mathrm{A}_{de} &= \frac{\text{volume passing in cubic feet per second} \times 144}{\text{velocity in feet per second}} \\ &= (258{\cdot}3 \times 144)/(62{\cdot}3 \times 52{\cdot}75) \\ &= 11{\cdot}3 \text{ sq. in.} \end{aligned}$$

The diameter of the diffuser throat is therefore 3·8 in.

The area A_{dd} at the discharge of the diffuser will be

$$\begin{aligned} \mathrm{A}_{dd} &= (11{\cdot}3 \times 52{\cdot}75)/18{\cdot}12 \\ &= 32{\cdot}89 \text{ sq. in.} \end{aligned}$$

The diameter at the discharge end is therefore 6·46 in.

The diameter or area of the exhaust inlet may be fixed by the same rules as were given on p. 57.

KÖRTING VACUUM BREAKER

The design of vacuum breaker fitted by Messrs. Körting to their ejector condensers is of the automatic type, that is, should, from any cause whatever, the water-level rise in the condenser to a predetermined and dangerous height the vacuum is automatically broken. The breaker is shown in fig. 47, and consists of a small cast-iron casing A which is bolted to the branch D shown in fig. 45 (p. 104). To this casing is fixed a lever and float S pivoted at B. A small relay valve C fitted to the casing A is operated by the rise and fall of the float S. The valve opens when the float rises, and closes when the float falls. The casing surrounding the valve C is carried up beyond the valve so as to form a water seal over the valve. The air admission valve V, to which the piston P is rigidly attached, is relay operated by the piston P through the medium of the relay valve C and the small communicating pipe D. The branch E is connected to the condenser.

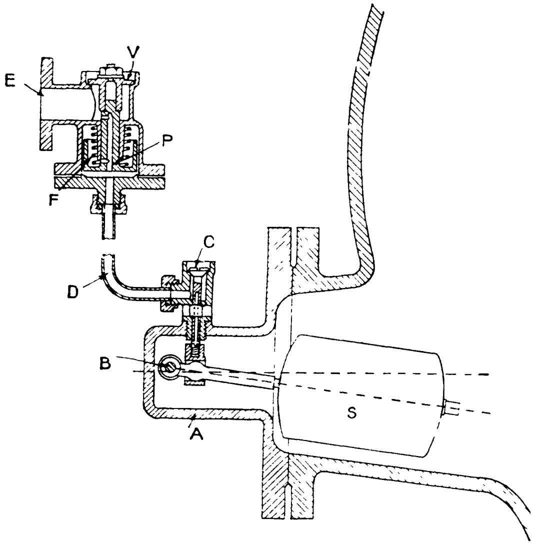


Fig. 47.—Vacuum Breaker fitted to the Körting Ejector Condensers

Under normal working conditions the float is in the lower position, the valves C and V are shut, and the chambers above and below the piston and the pipe D are all under the condenser vacuum, communication being effected by means of the small ducts through the stems of the valves C and V. The piston P therefore is in equilibrium, having the condenser vacuum above and below it. The valves V and C, having atmospheric pressure above them and the condenser vacuum below them, are therefore held shut. The closing force on the valve V is augmented by the pressure of the spring F. The action of the breaker is as follows.

When the float rises the valve C is opened and the air entering through this valve partially destroys the vacuum in the pipe D and under the piston P; the equilibrium of the piston is thus destroyed. There being a resultant upward thrust on the valve V and piston P the valve therefore opens and air flows in, and through the outlet branch E to the condenser, thus destroying the vacuum. When normal conditions are regained and the condenser is cleared of water the float falls, thus closing the valve C; the vacuum then rises in the pipe D, putting the piston in equilibrium, and the spring F closes the valve. The adoption of this automatic vacuum breaker does away with the necessity for a non-return valve in the exhaust pipe between the condenser and the engine or turbine, and avoids the inevitable drop in vacuum across such a valve due to its weight and friction.

These multi-jet ejector condensers must be erected vertically, and when starting up it is essential to have the condensing water circulating and the full vacuum showing in the condenser before admitting steam. When shutting down, the engine or turbine should be shut down first, and the pump supplying the condensing or injection water last.

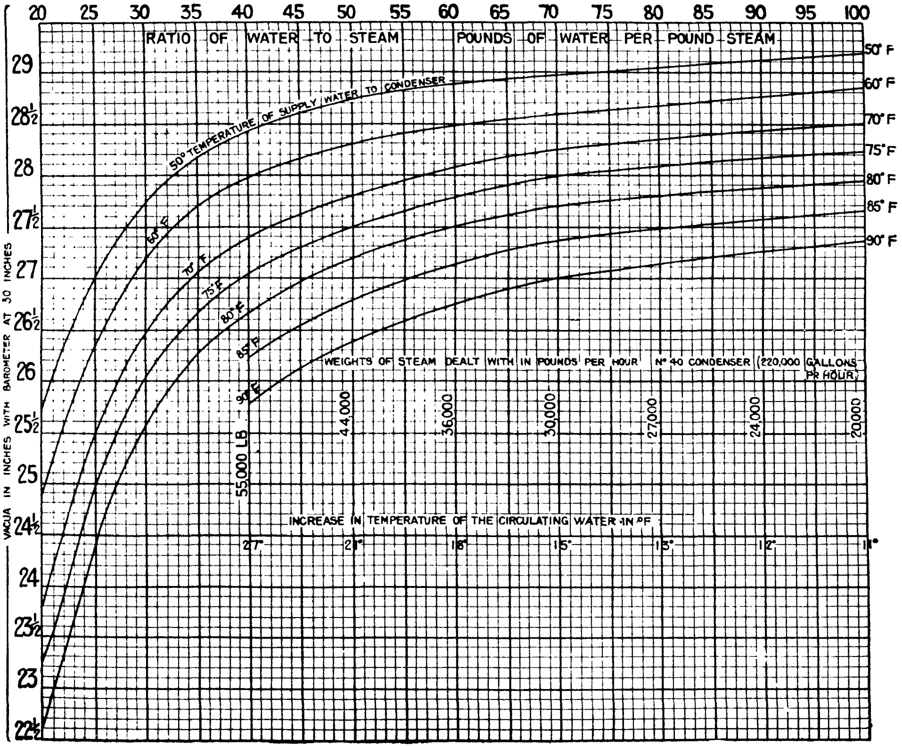


Fig. 48.—Test Results of a No. 40 Körting Ejector Condenser

Fig. 48 gives the results of a number of tests on a No. 40 Körting ejector condenser plotted as a series of curves. These curves were kindly supplied to the author by Messrs. Körting Brothers. During the tests the quantity of cooling water was maintained constant at 220,000 gall. per hour, and the steam quantity varied for each temperature of cooling water. A study of these curves is very instructive. At an inlet water temperature of 50° F. and a steam duty of 55,000 lb. per hour, the vacuum recorded was 28·45 in. Hg, while at the same inlet water temperature the vacuum at a steam duty of 44,000 lb. was 28·74 in. Hg. At the former steam duty the ratio of cooling water to steam was 40, while at the latter steam duty the ratio was 50, that is, for an increase in vacuum of 28·74 — 28·45 = 0·29 in. Hg, the water to steam ratio was increased 10 or 25 per cent. Again for a decreased steam duty from 44,000 to 27,000 lb. per hour the vacuum increase was 29·03 – 28·74 = 0·29 in. Hg, the same as before, and the water to steam ratio was increased to 80, an increase of 30 or 60 per cent; or in other words for an increase in vacuum of 0·29 in. Hg at about 28½ in. Hg. the quantity of cooling water must be increased by 25 per cent, the steam quantity remaining the same; while for an increase in vacuum of 0·29 in. Hg at about 28·9 in. Hg the quantity of cooling water must be increased 60 per cent, The same remarks apply to lower water temperatures, and show the large increase in the quantity of water necessary when vacua are maintained close to the vacua corresponding to the cooling water temperatures.

CHAPTER V

Evaporative Condensers

As its name implies, the evaporative condenser depends for its action on the rapid evaporation of a film of cold water flowing over the surface of a tube in the presence of air. The normal arrangement is to pass the steam through the inside of the tubes and a thin film of cooling water over the outside, at the same time allowing atmospheric air to circulate freely over the film of cooling water, either naturally or by means of fans.

Let W be the quantity of steam to be condensed in pounds per hour;
q, the dryness fraction of the steam;
G, the quantity of cooling water in pounds per hour;
L, the latent heat in the steam per pound;
t_2, the temperature of the condensate;
T_1, the inlet temperature of the cooling water;
T_2, the outlet temperature of the cooling water;
T, the temperature of the atmosphere;
G_e, the quantity of cooling water evaporated in pounds per hour;
t_1, the steam temperature; and
L_w, the average latent heat of vaporization of cooling water between T_1 and T_2.

The heat to be extracted from the steam per hour:

$$H_s = W(qL + t_1 - t_2).$$

Of the quantity of cooling water G entering the condenser the quantity G_e is evaporated and passes off with the air, and the quantity $(G - G_e)$ is heated from temperature T_1 to temperature T_2 and returned to the collecting tank to be recirculated. The quantity G_e is replaced by the introduction into the collecting tank of cold " make up " water, which mixing with the quantity $(G - G_e)$ maintains the inlet temperature at T_1.

The heat absorbed by the cooling water by evaporation will be

$$G_e L_w.$$

The average latent heat may be taken since the graph of latent heat and temperature is sensibly a straight line.

If Q_a be the quantity of air circulating in pounds per hour, and t_{1a} and

t_{2a} the inlet and outlet temperatures of the air, the heat absorbed by the air is

$$= Q_a(t_{2a} - t_{1a})r,$$

where r is the specific heat of the air.

The heat absorbed by the cooling water from temperature T_1 to T_2 is

$$(G - G_e)(T_2 - T_1).$$

Then

$$W(qL + t_1 - t_2) = (G - G_e)(T_2 - T_1) + Q_a(t_{2a} - t_{1a})r + G_eL_w. \dots (52)$$

Since the quantity $(G - G_e)$ is returned to the collecting tank at temperature T_2 and is cooled to temperature T_1 by the incoming quantity of " make up " water, G_e being heated from its inlet temperature T_{G_e} to temperature T_1, we get the following relation of a simple heat exchange:

$$(G - G_e)(T_2 - T_1) = G_e(T_1 - T_{G_e}) \dots\dots\dots\dots (53)$$

It should be noted that the temperature t_{2a} is the average temperature from top to bottom of the condenser.

The quantity of air necessary to evaporate G_e lb. of water depends upon the humidity of the entering and leaving air. The humidity of the entering air varies from day to day during the operation of any plant, and the value allowed for in the design is solely based on experience with these condensers. Normal atmospheric air, however, has a relative humidity of $\frac{2}{3}$, and this figure may be taken. The aim of the design of the cooling elements is to have the leaving air as completely saturated with vapour as possible, that is, to have the relative humidity of the leaving air unity.

Let R_1 be the relative humidity of the entering air;
R_2, the relative humidity of the leaving air;
w_1, the saturation weight of vapour associated with 1 lb. of air at temperature t_{1a}; and
w_2, the saturation weight of vapour associated with 1 lb. of air at temperature t_{2a}.

Then the weight of associated vapour entering the condenser per pound of dry air is w_1R_1, and the weight of associated vapour leaving the condenser is w_2R_2.

The weight of vapour taken up per pound of air will therefore be

$$w_2R_2 - w_1R_1,$$

and the total weight of air necessary to absorb G_e lb. of vapour will be

$$= G_e/(w_2R_2 - w_1R_1) \dots\dots\dots\dots\dots\dots (54)$$

Taking normal saturation of the entering air, R_1 will be $\frac{2}{3}$, and complete saturation of the leaving air $R_2 = 1$, we get:

$$Q_a = G_e/(w_2 - \tfrac{2}{3}w_1) \dots\dots\dots\dots\dots\dots (55)$$

To find w_1 and w_2 we know from Dalton's laws of partial pressures (p. 12) that the air and vapour both occupy the same volume and that each exerts its own pressure. Now 1 lb. of dry air at atmospheric pressure (14·7 lb. per square inch) and a temperature of 32° F. occupies 12·39 c. ft., so that

$$(14{\cdot}7 \times 12{\cdot}39)/(459{\cdot}6 + 32) = \mathrm{PV/T}. \quad \ldots\ldots\ldots\ldots (56)$$

Therefore, for any given pressure P and temperature T we can find the volume per pound. This volume will be occupied by the vapour at its temperature and partial pressure; and, since the specific volume can be found from the steam tables, the weight of vapour in the volume V can be found by dividing V by the specific volume. A concrete example will make this clear.

Example.—An evaporative condenser is to condense 10,000 lb. per hour of exhaust steam and maintain a vacuum of 25 in. Hg. The inlet and outlet temperatures of the circulating water are 90° F. and 110° F. respectively, and the temperature of the " make up " water is 50° F. The air temperature is 50° F., and the air has normal saturation. Calculate the quantity of cooling water entering the condenser and the weight of air circulating among the tubes. Assume the dryness fraction of the exhaust steam as 0·85, and the condensate temperature as 120° F., and the temperature of the leaving air and vapour as 90° F.

From equation (52) (p. 114) we have

$$\mathrm{W}(q\mathrm{L} + t_1 - t_2) = \mathrm{G}_e\mathrm{L}_w + (\mathrm{G} - \mathrm{G}_e)(\mathrm{T}_2 - \mathrm{T}_1) + \mathrm{Q}_a(t_{2a} - t_{1a})r.$$

q is 0·85;
L, 1016·5 B.Th.U.;
t_1, 134° F.;
t_2, 120° F.·
G_e is unknown;
L_w, (1041·2 + 1030)/2 = 1035·6 B.Th.U.;
W, 10,000 lb. per hour;
G is unknown;
T_1, 90° F.;
T_2, 110° F.;
r, 0·2375;
$\mathrm{T}_{\mathrm{G}_e}$, 50° F.;
Q_a is unknown;
t_{2a}, 90° F.; and
t_{1a}, 50° F.

From equation (53) (p. 114) we have:

$$\begin{aligned}(\mathrm{G} - \mathrm{G}_e)(\mathrm{T}_2 - \mathrm{T}_1) &= \mathrm{G}_e(\mathrm{T}_1 - \mathrm{T}_{\mathrm{G}_e}),\\ (\mathrm{G} - \mathrm{G}_e)(110 - 90) &= \mathrm{G}_e(90 - 50),\\ \text{or} \qquad 20\mathrm{G} - 20\mathrm{G}_e &= 40\mathrm{G}_e,\\ \mathrm{G}_e &= 20\mathrm{G}/60,\\ &= \mathrm{G}/3.\end{aligned}$$

From equation (55) we have:

$$Q_a = G_e/(w_2 - \tfrac{2}{3}w_1).$$

To find w_1 and w_2 we have the partial air pressure at the inlet temperature of 50° F. as:

$$\begin{aligned} p_{1a} &= 14{\cdot}7 - \tfrac{2}{3}\ \text{(the vapour pressure at } 50^\circ \text{ F.)} \\ &= 14{\cdot}7 - (2 \times 0{\cdot}178)/3 \\ &= 14{\cdot}58 \text{ lb. per square inch.} \end{aligned}$$

Then from equation (56),

$$\begin{aligned} &(14{\cdot}7 \times 12{\cdot}39)/(459{\cdot}6 + 32) = (14{\cdot}58V_{1a})/(459{\cdot}6 + 50), \\ &V_{1a} = (14{\cdot}7 \times 12{\cdot}39 \times 509{\cdot}6)/(14{\cdot}58 \times 491{\cdot}6) \\ &\quad\;\; = 12{\cdot}93 \text{ c. ft.} \end{aligned}$$

The specific volume of vapour at 50° F. from the steam tables is 1702 c. ft. per pound.

$$\begin{aligned} \therefore w_1 &= 12{\cdot}93/1702 \\ &= 0{\cdot}0076 \text{ lb. per pound of air.} \end{aligned}$$

Again, the partial air pressure at the outlet temperature of 90° F. is

$$\begin{aligned} &= 14{\cdot}7 - \text{vapour pressure at } 90^\circ \text{ F.} \\ &= 14{\cdot}7 - 0{\cdot}7 \\ &= 14 \text{ lb. per square inch.} \end{aligned}$$

Then from equation (56),

$$\begin{aligned} (14{\cdot}7 \times 12{\cdot}39)/(459{\cdot}6 + 32) &= 14V_{2a}/(459{\cdot}6 + 90), \\ \therefore V_{2a} &= (14{\cdot}7 \times 12{\cdot}39 \times 549{\cdot}6)/(14 \times 491{\cdot}6) \\ &= 14{\cdot}54 \text{ c. ft. per pound.} \end{aligned}$$

The specific volume of vapour at 90° F. is 469·3 c. ft.

$$\begin{aligned} \therefore w_2 &= 14{\cdot}54/469{\cdot}3 \\ &= 0{\cdot}0293 \text{ lb. per pound of air.} \end{aligned}$$

Then by substituting these values in equation (55) we get:

$$\begin{aligned} Q_a &= G_e/(w_2 - \tfrac{2}{3}w_1) \\ &= \tfrac{1}{3}G/(0{\cdot}0293 - \tfrac{2}{3} \times 0{\cdot}0076 \\ &= G/0{\cdot}0727, \end{aligned}$$

and, by substituting this value of Q_a and the value of $G_e = G/3$ in equation (52), we get

$$\begin{aligned} &W(qL + t_1 - t_2) \\ &\quad = \tfrac{1}{3}GL_w + (G - \tfrac{1}{3}G)(T_2 - T_1) + [\{G(t_{2a} - t_{1a})0{\cdot}2375\}/0{\cdot}0727], \end{aligned}$$

$$\begin{aligned} &\text{i.e. } 10{,}000(0{\cdot}85 \times 1016{\cdot}5 + 134 - 120) \\ &\quad = \tfrac{1}{3}G1035{\cdot}6 + \tfrac{2}{3}G(110 - 90) + [\{G(90 - 50)0{\cdot}2375\}/0{\cdot}0727], \\ &\text{i.e.}\qquad 8{,}780{,}000 = 345{\cdot}2G + 13{\cdot}33G + 130{\cdot}67G, \\ &\qquad\qquad \therefore G = 17{,}950 \text{ lb. per hour.} \end{aligned}$$

The weight of the air circulating will be

$$\begin{aligned} Q_a &= G/0{\cdot}0727 \\ &= 17{,}950/0{\cdot}0727 \\ &= 246{,}900 \text{ lb. per hour.} \end{aligned}$$

The volume occupied by this weight of air entering

$$\begin{aligned} &= 246{,}900 \times 12{\cdot}93 \\ &= 3{,}193{,}000 \text{ c. ft. per hour.} \end{aligned}$$

LEDWARD & BECKETT'S EVAPORATIVE CONDENSER

The evaporative condenser manufactured by Messrs. Ledward & Beckett of London is shown in fig. 49. The exhaust steam from the engines passes first through a grease separator and from there it passes to the exhaust main; this exhaust main runs, as shown, the whole length of the condenser, and from the underside head pipes lead the exhaust steam to the condensing elements. These condensing elements consist of corrugated and fluted cast-iron tubes bolted together and arranged in coils by means of suitable bends at the ends; the arrangement being such that the steam enters at the top of the element and flows backwards and forwards, the steam meantime condensing, till when the bottom flow or pass is reached the steam is all condensed. The bottom flows of the condensing elements are connected together by a condensate main which is connected to the air pump suction; the air pump discharges as usual to the hotwell. The circulating pump draws the circulating water from a large collecting tank, which is arranged underneath the condensing elements so that the water falling from these elements is caught in the tank. The circulating pump discharges to the top of the condenser distributing pipes and spreaders arranged over the corrugated and fluted cast-iron condensing elements; these distributing pipes and spreaders cause the water to spread itself evenly over the condensing elements in a thin film. The water flows by gravity from the top to the bottom tube and ultimately falls into the collecting tank arranged underneath. Large air spaces are provided between the elements and between the tubes comprising the elements, so that the air can circulate freely around the tubes, thereby creating conditions favourable for rapid evaporation. The arrangement and design of the corrugated tubes is the result of many years' experience, and the makers claim that the water is spread over the whole surface of the tubes in a perfectly even and thin film. It will be readily understood that this is a very important point, since if portions of the tube surface were dry these portions would be ineffective for the condensation of steam, and the maximum efficiency would not be obtained. The quantity of water G_e lost by evaporation does not, with these condensers, exceed two-thirds the quantity of steam at the maximum load, while at lighter loads the quantity is considerably less. At a load of about 30 per cent of full load the water circulation can be stopped and the steam condensed by air cooling alone.

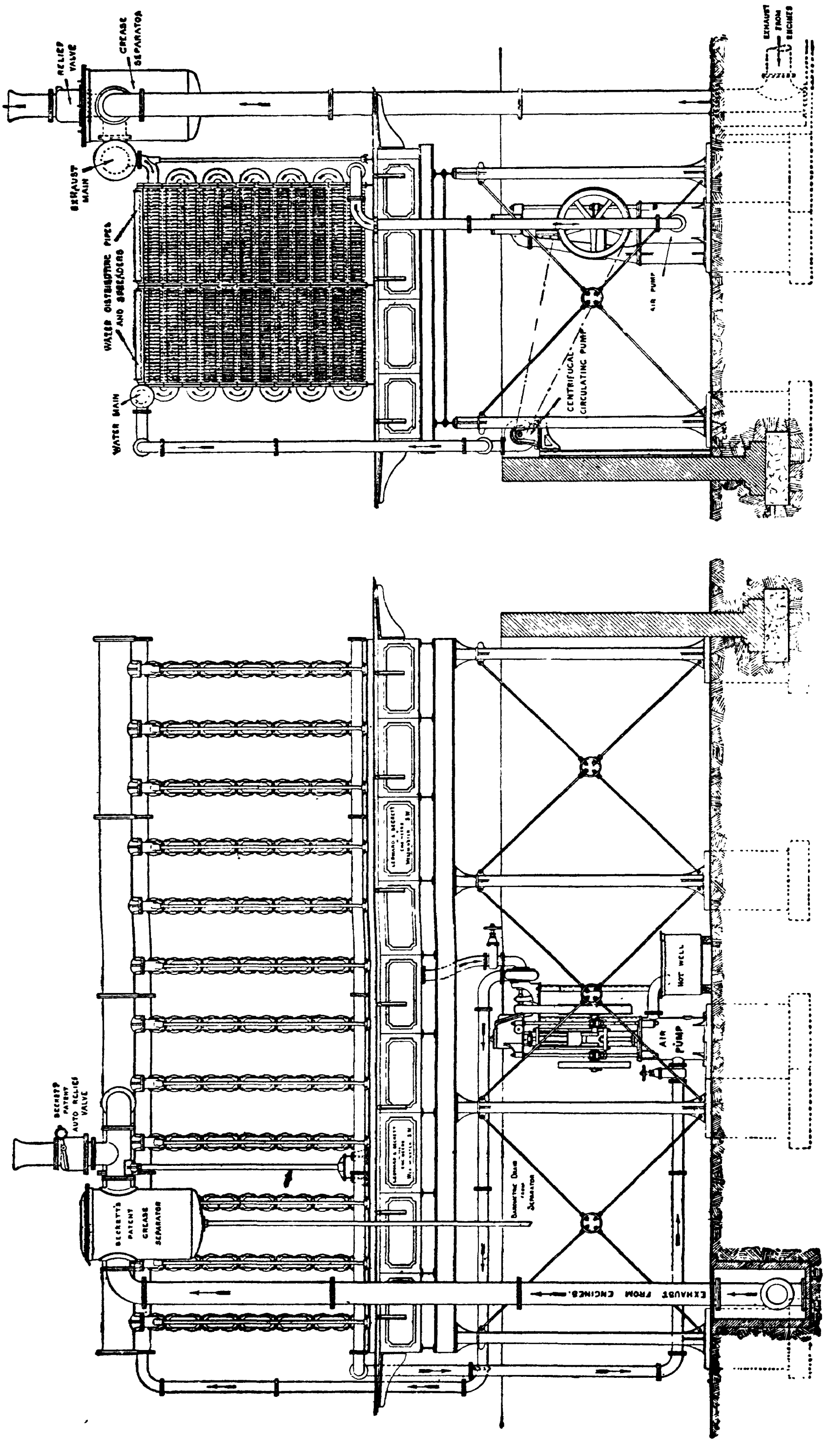

RELIEF VALVE
GREASE SEPARATOR
EXHAUST MAIN
WATER DISTRIBUTING PIPES AND SPREADERS
WATER MAIN
CENTRIFUGAL CIRCULATING PUMP
AIR PUMP
EXHAUST FROM ENGINES
BECKETT'S PATENT AUTO RELIEF VALVE
BECKETT'S PATENT GREASE SEPARATOR
BAROMETRIC DRAIN FROM SEPARATOR
HOT WELL
AIR PUMP
EXHAUST FROM ENGINES.

Fig. 49.—Ledward & Beckett Patent Evaporative Condenser

This is a valuable feature in works which carry a light load for considerable portions of the day or night, as the circulating pump can be shut down and a considerably economy effected thereby.

Since the steam and circulating water both enter at the top and gravitate towards the bottom, the condenser is of the parallel-flow type. At full loads this arrangement gives a condensate temperature approaching the vacuum temperature, but at light loads there may be a considerable depression of condensate temperature below the vacuum temperature. This depression is a direct loss, and to minimize this Messrs. Ledward & Beckett have recently arranged a new system of stage drainage whereby the condensate is trapped off from the coils by means of special baffle deflectors. This system is shown in fig. 50 which shows a typical plant of this description.

It will be noticed that from every alternate coil a drain pipe is led to a vertical drain collector fitted with an internal pipe. The communication between the internal pipe and the drain collector is through a water seal, so that the condensate drained off at each stage must pass through its own trap before reaching the internal pipe. Short circuiting of the steam is thereby prevented, and, since the condenser is hotter at the top than it is at the bottom, the condensate tapped off at the higher stages will be hotter than that tapped off lower down; the net result of the arrangement being an increased hotwell temperature for the same vacuum, which makes for economy.

Vacua of 24 in. Hg and upwards can be obtained with these condensers, and it is interesting to note that the running costs vary only very slightly for high or low vacuum; the normal vacuum, however, recommended is 25 in. Hg at full load, this being the figure found by the makers to be the best for true fuel economy.

A notable feature of this type of condenser is the low power consumption of the circulating pump. The normal quantity of water pumped is 1 gall. per pound of steam, or a ratio of water to steam of 10 to 1; the head pumped against is normally about 12 ft. 6 in., this being the difference in level between the water in the tank and the top of the condenser. The power consumption therefore works out at about 0·25 per cent of the power equivalent of the steam being condensed. The total power consumption for working the condenser varies between 2 and slightly under 1 per cent, according to the size of the plant; the former figure is for small plants and the latter figure for large plants.

Owing to its great volumetric capacity the evaporative condenser is unique in its being able to take heavy overloads for short periods without the vacuum being adversely affected, the condenser is thus eminently suited to engines which have heavy peak loads such as rolling mill engines, &c.

The following test data, kindly supplied to the author by Messrs. Ledward & Beckett from one of their evaporative condensers, shows a normal performance of their evaporative condensers.

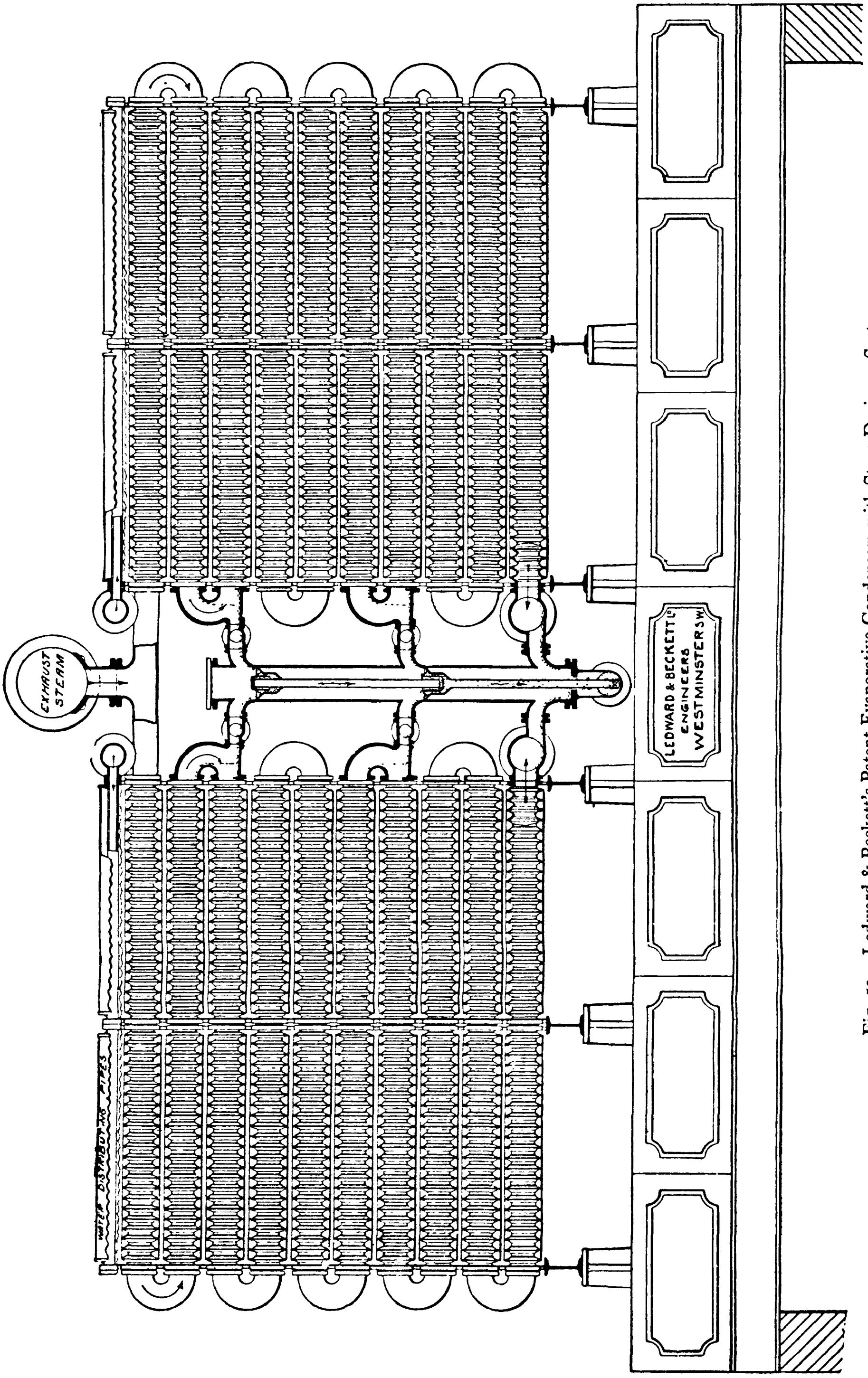


Fig. 50.—Ledward & Beckett's Patent Evaporative Condenser; with Stage Drainage System

Evaporative Condenser. (L. & B.)

Data received from the Broughton and Plas Power Coal Company, Ltd., near Wrexham.

Capacity of condenser	15,000 lb. per hour.
Vacuum	25 in. Hg.
Air pump	Steam driven.
Circulating pump	Motor driven.
Engines in use	One 600-kw. Belliss & Morcom, triple compound. One 200-kw. Scott & Mountain, vertical high-speed compound. One 200-kw. Roley-Horiz, compound.
Load	Fluctuating between 500 and 800 amp. at 550 volts.
Quantity of steam condensed	Average 12,000 lb. per hour.
Temperature of circulating water	Average 90° F.
Condensate	Average 110° F.
Atmospheric temperature	Average 50° F.
Barometer	Average 28·5 in. Hg.
Vacuum	Average 25·5 to 26·5 in. Hg.
Vacuum	Correct to barometer 27 to 28 in. Hg.
Average power consumed by circulating pump	3·5 b.h.p.
Steam used by pump	150 lb. per hour and heat recovered in condensate.
Amount of "make-up" introduced into the tank under condenser	800 gall. per hour.

Making the following analysis of the figures we find that the water lost by evaporation is

$$\begin{aligned} &= (8000 \times 100)/12{,}000 \\ &= 66{\cdot}6 \text{ per cent of the steam condensed.} \end{aligned}$$

The power consumption of the circulating pump is 3·5 b.h.p.

The power consumption of the air pump taking 30 lb. of steam per b.h.p., and neglecting any heat recovery is $150/30 = 5$ b.h.p.

Since no fans are used for air circulation by Messrs. Ledward & Beckett the total power consumption is 8·5 b.h.p., and taking an average steam consumption of 20 lb. per brake horse-power for the engines exhausting to the condenser, the percentage of power consumption to the power condensed is

$$\begin{aligned} &(8{\cdot}5 \times 20 \times 100)/12{,}000 \\ &= 1{\cdot}415 \text{ per cent.} \end{aligned}$$

EXAMPLES ON EVAPORATIVE CONDENSERS

Example 1.—On a certain day the temperature of the air supply to an evaporative condenser was 55° F. The dew-point as measured by the wet and dry bulb hygrometer was 40° F. Find the weight of vapour associated with each pound of air which was entering the condenser.

Answer.—0·00523 lb. per pound of air.

Example 2.—The temperature of the air and vapour leaving an evaporative condenser was measured to be 115° F. and the condition of the air entering was as given in example 1. Find the total quantity of air which must be in circulation to evaporate 4000 lb. of water per hour. Assume the air leaving the condenser to be saturated with water vapour.

Answer.—62,360 lb. per hour.

Example 3.—An evaporative condenser is to maintain a vacuum of 25 in. Hg, the supply of " make up " water is at a temperature of 65° F. The temperature of the condensate is 5° F. below the vacuum temperature, and the outlet temperature of the circulating water is 115° F. The quantity of exhaust steam to be dealt with is 9000 lb. per hour, and the steam has a dryness fraction of 0·9. The entering air temperature is 50° F., and the dew-point 40° F.

Calculate the minimum quantity of " make up " water introduced into the collecting tank to maintain the circulating water inlet temperature at 100° F.. and state the quantity of circulating water entering the condenser.

Answer.—6419 lb. per hour. The quantity of water entering is 21,397 lb. per hour.

CHAPTER VI

Air Ejectors and Air Pumps

STEAM-JET OPERATED AIR EJECTORS

Steam-jet operated air ejectors have come to be almost universally used for the production of the high vacuum demanded in the modern land condensing plant, owing to their simplicity, the small space occupied, and the absence of moving parts with the consequent reliability.

There are many makes now on the market, and they all follow the same general principles. The air is compressed in two stages, the first stage compressing the air from the condenser vacuum to 25 or 26 in. Hg vacuum, and the second stage compressing from this vacuum to atmospheric pressure. Both stages follow the same general principles, i.e. the potential energy of the operating steam is transformed in a convergent-divergent steam nozzle into velocity energy, so that the steam issues from this nozzle with the maximum velocity. This rapidly moving jet of steam entrains the air and non-condensible gases by friction, and the combined mass of entrained air and steam is discharged into a diffuser nozzle, where a portion of the kinetic energy is retransformed into pressure energy, so that the air is discharged against a pressure higher than the ejector suction pressure.

Let fig. 51 represent one stage of an air ejector.

Let v_s be the exit velocity of the steam jet in feet per second;
v_t, the velocity at the diffuser throat in feet per second;
v_d, the velocity at the diffuser outlet in feet per second;
v_a, the velocity of the air and vapour when it meets the issuing jet of steam in feet per second;
P_t, the absolute pressure in pounds per square inch at the diffuser throat;
P_d, the absolute pressure in pounds per square inch at the diffuser outlet;
p_a, the partial air pressure at entrance in pounds per square inch;
m, the mass of dry air in pounds per second;
m_v, the mass of vapour associated with m lb. of air;
M, the mass of steam flowing in pounds per second;
J, Joule's mechanical equivalent of heat;
E_d, the efficiency of the diffuser nozzle;
V_t, the volume per pound of mixture at the diffuser throat in cubic feet; and
V_d, the volume per pound of mixture at the diffuser discharge in cubic feet.

From the laws of impact or collision we know that the sum of the momenta before impact is equal to the sum of the momenta after impact, therefore:

$$Mv_s + v_a(m + m_v) = (M + m + m_v)v_t,$$

$$\text{or} \quad v_t = \{Mv_s + v_a(m + m_v)\}/(M + m + m_v) \ldots\ldots\ldots (57)$$

From this equation we can find the velocity v_t at the diffuser throat for given masses and velocities of steam, air, and vapour. In the above the assumption is made that there is no rise in pressure before the diffuser throat is reached. This is not strictly true however.

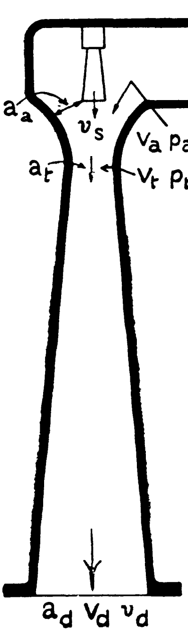


Fig. 51.—One Stage of an Air Ejector

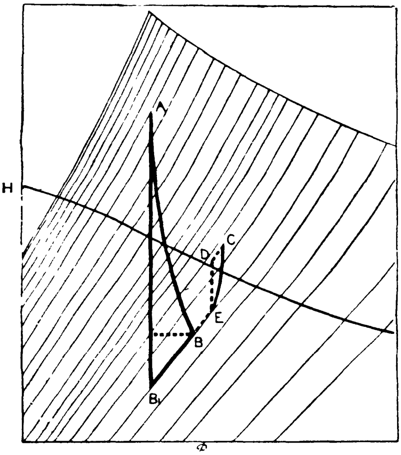


Fig. 52.—Total Heat Entropy Chart

The combined mass of air, vapour and steam flowing past the throat section with a velocity v_t is gradually reduced in velocity and increased in pressure in the diverging part of the diffuser.

Assuming that the compression is a reversed expansion, the steam conditions may be shown on the total heat entropy chart, fig. 52.

Point A gives the steam supply conditions to the ejector, expansion takes place from A to B in the steam nozzle, point B giving the condition at the steam nozzle exit. The drop in total heat from A to B equals the gain in velocity energy of the steam or, neglecting the small initial velocity of the steam,

$$J(H_A - H_B) = v_s^2/(2g) \ldots\ldots\ldots\ldots\ldots\ldots (58)$$

Between the points B and E entrainment of the air and vapour takes place. The dryness of the steam will therefore be slightly increased and there will be an increase in entropy along a constant pressure line. The velocity v_s at B will drop to v_t at E. The mass of steam flowing will now be increased by the mass of entrained vapour. Taking 1 lb. of this combined mass of steam and entrained vapour the pressure rise in the diffuser nozzle will be from E to C, the velocity v_t at E having meanwhile fallen to v_d at C. The pressure at C will be less than the discharge pressure by the amount of the air pressure. If the compression had been frictionless and adiabatic the compression line would have been ED, and the discharge velocity would have been increased. Owing to friction, however, entropy is added to the steam, and the compression line takes the form EC, the action being similar to the expansion in a steam nozzle where the reheat increases the entropy and causes the truly adiabatic to move from AB′ to AB.

The combined mass of air, vapour, and steam is compressed along the line EC and if n be the compression exponent of the mixture, and n_s the compression exponent of the pure steam, and n_a the compression exponent of the pure air, the following relation holds:

$$p/(n-1) = p_a/(n_a-1) + p_s/(n_s-1), \quad \ldots\ldots\ldots\ldots(59)$$

where p is the total absolute pressure at any point in the compression,
p_a is the air pressure at the same point, and
p_s is the steam pressure at the same point.

The value of n_a for pure dry air is 1·4, and the value of n_s, which would cause the discharge to be just dry, would be Zeuner's value for dry steam, namely, 1·135. Owing, however, to the fact that steam does not readily pick up moisture during compression, the exponent n_s is nearer 1·3 than 1·135 and may be taken at the former value. The influence of this factor, and the frittering down into heat of the friction losses in the diffuser, are the main causes for the discharge from air ejectors of the reversed nozzle type being slightly superheated, even when the ejector is supplied with wet steam.

If a thermometer be placed in the discharge from an air ejector discharging to atmosphere, the temperature indicated may be lower than 212° F. This does not necessarily indicate that the discharge is wet. The total pressure at discharge is the sum of the air pressure and the steam pressure; and, if the air quantity passing is large, the air pressure will be large and the steam pressure low, so that the saturation temperature of the steam will be less than 212° F. In general it will be found that the larger the air quantity passing, the lower will be the discharge temperature. This temperature may vary from 220° F. to 205° F. in the same ejector, the variation being obtained by varying the air quantity passing.

The available energy supplied at the throat of the diffuser nozzle is $v^2_t/(2g)$ per pound of mixture. The work done per pound in compressing and discharging the mixture is

$$\{n/(n-1)\}(P_dV_d - P_tV_t)144, \quad \ldots\ldots\ldots\ldots\ldots(60)$$

and the residual energy at the diffuser discharge is

$$v^2_d/(2g).$$

The available energy will equal the work done plus the residual energy, i.e. $v^2_t/(2g) = \{n/(n-1)\}(P_dV_d - P_tV_t)144 + v^2_d/(2g)$. ....(61)

Allowing for the efficiency of the diffuser this expression becomes

$$E_dv^2_t/(2g) = \{n/(n-1)\}(P_dV_d - P_tV_t)144 + v_d^2/(2g)\text{.....}(62)$$

The compression will follow the law

$$PV^n = \text{constant.}$$

Again, let a_a be the annular area in square inches between the exit end of the steam nozzle and the diffuser cone;

a_t, the area in square inches of the diffuser throat;

a_d, the area in square inches of the diffuser outlet; and

V_a, the volume in cubic feet per pound of air at the partial air pressure p_a at the area a_a;

then the entrance area a_a is

$$a_a = (144V_am)/v_a\text{. ....(63)}$$

At the throat section the necessary area a_t must be such as to allow the combined mass of steam, air, and vapour to pass at the velocity v_t. If V_t be the volume per pound of the mixture and q the dryness fraction of the steam issuing from the steam nozzle, then

$$V_t = (M \times \text{specific volume} \times q + mV_a)/(M + m + m_v)\text{...(64)}$$

The throat area a_t will be

$$a_t = \{144V_t(M + m + m_v)\}/v_t\text{. ....(65)}$$

Similarly at the diffuser discharge the necessary area a_d will be

$$a_d = \{144V_d(M + m + m_v)\}/v_d\text{, ...(66)}$$

where V_d is the volume in cubic feet per pound of mixture at the discharge pressure P_d.

Example.—A single-stage air ejector is to withdraw 40 lb. of dry air per hour saturated with water vapour at a temperature of 80° F. from a vessel at a vacuum of 26 in. Hg. The exit velocity from the steam nozzle is 3495 ft. per second, and the dryness fraction of the steam is 0·846. The mass of steam flowing is 392 lb. per hour; calculate the necessary areas: a_t, a_a, and a_d. Take the velocity v_a = 100 ft. per second.

First find the mass of vapour m_v which is associated with m lb. of air at 26 in. Hg vacuum and 80° F.

The partial air pressure p_a will be

$$\begin{aligned} p_a &= 28{\cdot}97 - 26 \\ &= 2{\cdot}97 \text{ in. Hg,} \\ \text{or} \quad &= (2{\cdot}97 \times 14{\cdot}7)/30 \\ &= 1{\cdot}455 \text{ lb. per square inch.} \end{aligned}$$

The volume per pound of air at this partial pressure will be

$$\begin{aligned} V_a &= [(12{\cdot}39 \times 30)(459{\cdot}6 + 80)]/[2{\cdot}97(459{\cdot}6 + 32)] \\ &= 137{\cdot}4 \text{ c. ft. per pound.} \end{aligned}$$

The specific volume of the vapour at 80° F. is from the steam tables 636·8 c. ft. per pound.

The mass of vapour associated with 1 lb. of air will then be

$$137{\cdot}4/636{\cdot}8 = 0{\cdot}2157 \text{ lb. per pound,}$$

and m_v will be

$$\begin{aligned} m_v &= (0{\cdot}2157 \times 40)/3600 \\ &= 0{\cdot}00239 \text{ lb. per second.} \end{aligned}$$

The total mass of air and vapour entering the ejector suction will be, say m_a,

$$\begin{aligned} m_a &= m + m_v \\ &= (40/3600) + 0{\cdot}00239 \\ &= 0{\cdot}0135 \text{ lb. per second.} \end{aligned}$$

From equation (63) the area a_a will be

$$\begin{aligned} a_a &= (144V_a m)/v_a \\ &= (144 \times 137{\cdot}4 \times 40)/(100 \times 3600) \\ &= 2{\cdot}2 \text{ sq. in.} \end{aligned}$$

Again, from equation (57) the throat velocity v_t is

$$\begin{aligned} v_t &= \{Mv_s + v_a(m + m_v)\}/(M + m + m_v) \\ &= [\{(392 \times 3495)/3600\} + 100 \times 0{\cdot}0135]/[(392/3600) + 0{\cdot}0135] \\ &= 3121 \text{ ft. per second.} \end{aligned}$$

From the equation (64) V_t will be

$$\begin{aligned} V_t &= (M \times \text{specific volume} \times q + mV_a)/(M + m + m_v) \\ &= (0{\cdot}1089 \times 175 \times 0{\cdot}846 + 0{\cdot}01111 \times 137{\cdot}4)/(0{\cdot}1089 + 0{\cdot}01111 + 0{\cdot}00239) \\ &= 144 \text{ c. ft.} \end{aligned}$$

Then substituting these values in equation (65) we get

$$\begin{aligned} a_t &= 144V_t(M + m + m_v)/v_t \\ &= (144 \times 144 \times 0{\cdot}1224)/3121 \\ &= 0{\cdot}8164 \text{ sq. in.} \end{aligned}$$

The diameter having this area is $1\frac{1}{64}$ in.

From equation (59) the compression exponent n may be found:

$$p/(n - 1) = p_a/(n_a - 1) - p_s/(n_s - 1),$$

$$n_a = 1{\cdot}4, \text{ and } n_s = 1{\cdot}3.$$

At the throat section:

$$\begin{aligned} p_a &= 1{\cdot}455 \text{ lb. per square inch,} \\ p_t &= 4 \text{ in. Hg or } 1{\cdot}96 \text{ lb. per square inch, and} \\ p_s &= 1{\cdot}96 - 1{\cdot}455 = 0{\cdot}505 \text{ lb. per square inch.} \end{aligned}$$

Then substituting we get

$$\begin{aligned} 1{\cdot}96/(n-1) &= 1{\cdot}455/(1{\cdot}4-1) + 0{\cdot}505/(1{\cdot}3-1), \\ \therefore n &= 1{\cdot}368. \end{aligned}$$

Again we have:

$$\begin{aligned} \mathrm{PV}^n &= \mathrm{C}, \\ \text{or} \qquad \mathrm{P}_t\mathrm{V}_t^{1\cdot368} &= \mathrm{P}_d\mathrm{V}_d^{1\cdot368}, \\ \mathrm{V}_d &= \mathrm{V}_t(\mathrm{P}_t/\mathrm{P}_d)^{1/1\cdot368} \\ &= 144(1{\cdot}96/14{\cdot}7)^{1/1\cdot368} \\ &= 144 \times 0{\cdot}2294 \\ &= 33 \text{ c.ft.} \end{aligned}$$

From equation (62):

$$\mathrm{E}_d v_t^2/(2g) = \{n/(n-1)\}(\mathrm{P}_d\mathrm{V}_d - \mathrm{P}_t\mathrm{V}_t)144 + v_d^2/(2g).$$

Taking the efficiency of the diffuser at 72 per cent we get

$$\begin{aligned} 0{\cdot}72 \times (3121)^2/(2g) & \\ &= \{1{\cdot}368/(1{\cdot}368-1)\}(14{\cdot}7 \times 33 - 1{\cdot}96 \times 144)144 + v_d^2/(2g), \\ 108{,}900 &= 108{,}106 + v_d^2/(2g), \\ \text{or} \qquad v_d &= \sqrt{51{,}133} \\ &= 226{\cdot}1 \text{ feet per second.} \end{aligned}$$

It may be instructive to work out the diffuser efficiency for this case at which the ejector would break down. This point is reached when the discharge velocity V_d becomes zero, or when

$$\mathrm{E}_d v_t^2/(2g) = \{n/(n-1)\}(\mathrm{P}_d\mathrm{V}_d - \mathrm{P}_t\mathrm{V}_t)144.$$

By substitution we find E_d to be

$$\begin{aligned} \mathrm{E}_d &= 108{,}106/151{,}252 \\ &= 0{\cdot}7147, \\ \text{or} \qquad & 71{\cdot}47 \text{ per cent.} \end{aligned}$$

The discharge area a_d would be, from equation (66),

$$\begin{aligned} a_d &= \{144\mathrm{V}_d/v_d\}(\mathrm{M} + m + m_v) \\ &= \{(144 \times 33)/226{\cdot}1\}0{\cdot}1224 \\ &= 2{\cdot}57 \text{ sq. in.} \end{aligned}$$

The diameter which has this area is $1\frac{13}{16}$ in.

Normally ejectors are designed to discharge against a small back pressure to overcome the friction loss in the discharge pipe, and to reduce this loss to a minimum the velocity in the discharge pipe is kept low. If the above ejector therefore had to discharge through a discharge pipe the exit diameter of $1\frac{13}{16}$ in., as found above, would be increased to keep the discharge velocity under 100 ft. per second.

STEAM NOZZLES

The steam nozzles adopted in steam-jet operated air ejectors are of the convergent-divergent type and are of normal design. The expansion ratio is, however, very large, since they discharge directly into vacua.

The author does not propose to deal largely with the theory of steam nozzles, since the theory is dealt with very fully in many textbooks on steam turbines. (See *Steam Turbines* by Goudie.) He proposes rather to give the generally accepted formulæ for exit velocity, &c., so far as they are applicable to steam-jet air ejectors.

Let P be the initial steam pressure in pounds per square inch absolute;
V, the initial volume of the steam in cubic feet per pound;
p, the pressure at exit in pounds per square inch absolute;
v_s, the exit steam velocity in feet per second;
v_t, the throat velocity in feet per second;
P_t, the throat pressure in pounds per square inch absolute;
w, the weight discharged in pounds per second;
E, the efficiency of the nozzle;
a, the area of the throat in square inches;
A, the area of the throat exit in square inches;
V_0, the actual volume of steam at exit in cubic feet per pound;
V_s, the volume of dry steam at exit in cubic feet per pound;
q, the quality of the steam at exit;
h, the heat drop from P to p.

For initially dry saturated steam and for superheated steam,

$$a = (w/0{\cdot}3155)\sqrt{V/P} \quad \ldots\ldots\ldots\ldots\ldots\ldots (67)$$

It can be shown that

$$P_t/P = 0{\cdot}5457,$$

and that

$$v_t = 72{\cdot}24\sqrt{PV} \quad \ldots\ldots\ldots\ldots\ldots\ldots (68)$$

It should be noted that the numerical value of V_s will depend on the degree of superheat; the value may be found directly from the steam tables.

The exit velocity of the steam is calculated from

$$v_s = 223{\cdot}7\sqrt{Eh} \quad \ldots\ldots\ldots\ldots\ldots\ldots (69)$$

The heat drop h is best read directly from the total heat entropy chart, and the value of E may be deduced from the following equation, which is due to H. M. Martin, who obtained it by collation of the results of various experimenters:

$$100\mathrm{E} = 102{\cdot}7 - 0{\cdot}06h \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots(70)$$

The exit volume of the steam is given by

$$\mathrm{V}_0 = q\mathrm{V}_s \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(71)$$

The area of the nozzle exit is found from

$$\mathrm{A} = 144w\mathrm{V}_0/v_s \quad\ldots\ldots\ldots\ldots\ldots\ldots(72)$$

Example.—The second stage steam nozzle of an air ejector is to pass 392 ib. per hour. The steam supply pressure is 150 lb. per square inch gauge, and the steam is dry saturated. The intermediate vacuum of the ejector is 26 in. Hg. Calculate the necessary throat diameter and the necessary exit diameter of the steam nozzle, and the exit velocity v_s. State what the dryness fraction of the steam issuing from the steam nozzle will be.

From equation (67):

$$\begin{aligned} a &= (w/0{\cdot}3155)/\sqrt{\mathrm{V}/\mathrm{P}} \\ &= \{392/(3600 \times 0{\cdot}3155)\}/\sqrt{2{\cdot}758/164{\cdot}7} \\ &= 0{\cdot}04446 \text{ sq. in.} \end{aligned}$$

The equivalent diameter is 15/64 in. (nearly).

From equation (70):

$$100\mathrm{E} = 102{\cdot}7 - 0{\cdot}06h,$$

and from the total heat entropy chart h from P to p, i.e. from 165 lb. per square inch absolute to 26 in. Hg vacuum is 289 B.Th.U.

$$\begin{aligned} \therefore\ 100\mathrm{E} &= 102{\cdot}7 - 0{\cdot}06 \times 289 \\ &= 102{\cdot}7 - 17{\cdot}34, \\ \text{i.e. } \mathrm{E} &= 85{\cdot}36 \text{ per cent.} \end{aligned}$$

Then 85·36 per cent of 289 = 246·8 B.Th.U., and scaling this on the total heat entropy chart gives $q = 0{\cdot}82$.

From equation (69):

$$\begin{aligned} v_s &= 223{\cdot}7\sqrt{\mathrm{E}h} \\ &= 223{\cdot}7\sqrt{0{\cdot}8534 \times 289} \\ &= 3512 \text{ ft. per second.} \end{aligned}$$

From equation (71) the actual volume at exit V_0 is

$$\begin{aligned} \mathrm{V}_0 &= q \times \mathrm{V}_s \\ &= 0{\cdot}82 \times 175 \\ &= 143{\cdot}5 \text{ c ft. per pound.} \end{aligned}$$

From equation (72) the exit area A is

$$\begin{aligned} A &= 144wV_0/v_s \\ &= (392 \times 143{\cdot}5 \times 144)/(3600 \times 3512) \\ &= 0{\cdot}6406 \text{ sq. in.,} \end{aligned}$$

i.e. the exit diameter of ejector is 29/32 in

In the equation of constant energy (equation 61) v_d must be a positive quantity. If v_d were zero or negative, the stream in the diffuser nozzle would break and the ejector would become deprimed. This condition of instability occurs if the diffuser throat area and discharge area are not properly proportioned to the quantity and velocity of the steam issuing from the steam nozzle and the quantity of air passing. In other words, if an ejector is designed to deal with a large quantity of air at a relatively low vacuum, and if the vacuum be raised by cutting down the air quantity, the ejector would become unstable at the higher vacuum unless the steam pressure be increased, so that the discharge velocity v_d is maintained a positive quantity.

By allowing a variation in the discharge velocity v_d an air ejector can be stably operated, when dealing with the relatively small air quantities, if the ratio between the discharge pressure and the suction pressure, or the compression ratio, as it is called, be kept under 8 to 1. Now the ratio between atmospheric pressure and a vacuum of 29·5 in. Hg is 60 to 1, but if the compression is performed in two stages, the intermediate absolute pressure p for this overall compression ratio would be given by

$$\begin{aligned} p/0{\cdot}5 &= 30/p, \\ \text{i.e. } p^2 &= 15, \\ p &= 3{\cdot}87 \text{ in. of mercury absolute,} \end{aligned}$$

and the compression ratio on each stage would be

$$3{\cdot}87/0{\cdot}5 = 30/3{\cdot}87 = 7{\cdot}74.$$

The intermediate vacuum would be

$$(30 - 3{\cdot}87) = 26{\cdot}13 \text{ in. Hg.}$$

A two-stage air ejector, therefore, will operate quite stably when dealing with varying quantities of air, and easily maintain a vacuum up to 29·5 in. Hg, when the conditions of temperature permit. It is on this question of stability that the majority of steam-jet operated air ejectors on the market to-day differ. They all employ two-stage compression with convergent-divergent steam nozzles, but vary in the means adopted to ensure stability. Monsieur Le Blanc, who did much of the pioneer work of steam-jet operated air ejectors, adopted the plan of admitting air into the throat of the second-stage diffuser; this virtually gave a throat diameter which varied with the quantity of air being dealt with, and this varied the virtual ratio of throat area to exit area.

Monsieur Delaporte adopted the plan of admitting air between the stages of his ejectors; this air he admitted through a spring loaded valve which came into operation when the vacuum between the stages or the intermediate vacuum exceeded 26 in. Hg. This plan ensured that the compression ratio of the second stage would never go beyond 8 to 1, so that stability of this stage would be maintained and the first stage could then operate stably up to 29·5 in. Hg.

The Delas air ejector maintains stability by means of water jacketing the diffuser nozzles. This water jacketing gives virtually a varying throat diameter, since at low vacua or when air leakage is present the diffuser is hotter than it is at high vacua or when the air leakage is small; a greater quantity of the steam passing through the diffuser will therefore be condensed at low vacua than at high vacua, owing to the greater temperature drop across the diffuser wall, and, consequently, the virtual throat diameter will be larger at low vacua than at high vacua, thus providing sufficient area for the greater air quantity passing at the low vacua.

Many modern steam-jet operated air ejectors, however, simply rely on the variation in the discharge velocity v_d and air dealing efficiency over the working range of air leakage. Ejectors designed on this principle will, however, have a smaller air dealing capacity at abnormally large air leakages, but against this the ejector is simplified, and on any specific plant it should always be possible to maintain the leakage under the allowable value given on p. 37.

The following relation is sometimes used to estimate the air and vapour extracted per pound of steam used by an air ejector of the reversed nozzle type. It is based on the following assumptions:

1. That the compression exponent in the diffuser nozzle is the same as the expansion exponent in the steam nozzle, and that the expansion and compression are adiabatic.
2. That the discharge velocity v_d is zero. The work done per pound of steam in the nozzle in imparting velocity to the steam is

$$\{n/(n-1)\}(PV - P_1V_1)144,$$

where P_1 is the suction pressure in pounds per square inch absolute, and
V_1 is the suction volume in cubic feet per pound of steam.
If w is the weight of air and vapour extracted per pound of steam used, i.e. $(m + m_v)$;
P_d, the discharge pressure in pounds per square inch absolute; and
V_d, the volume at discharge in cubic feet per pound of mixture;
then the total work done in the diffuser per pound of steam used is

$$\{n/(n-1)\}(P_dV_d - P_1V_1)(w+1)144.$$

These two quantities must be equal,

$$\therefore \{n/(n-1)\}(PV - P_1V_1)144 = \{n/(n-1)\}(P_dV_d - P_1V_1)(w+1)144.$$

Again, if E_d be the diffuser efficiency, and E the efficiency of the steam nozzle, the above expression becomes:

$$\{n/(n-1)\}(PV-P_1V_1)144E = \{n/(n-1)\}(P_dV_d-P_1V_1)(w+1)144\times 1/E_d,$$

$$\text{or} \qquad w = EE_d[(PV-P_1V_1)/(P_dV_d-P_1V_1)]-1.$$

Since the expansion and compression are adiabatic:

$$PV^n = P_1V_1^n,$$

$$\text{or } V = V_1(P_1/P)^{1/n},$$

$$\text{and } V_d = V_1(P_1/P_d)^{1/n}.$$

Then, substituting these values of V and V_d, we get

$$w = EE_d[\{PV_1(P_1/P)^{1/n} - P_1V_1\}/\{P_dV_1(P_1/P_d)^{1/n} - P_1V_1\}] - 1$$

$$= EE_d[\{P^{(n-1)/n} - P_1^{(n-1)/n}\}/\{P_d^{(n-1)/n} - P_1^{(n-1)/n}\}] - 1. \quad \ldots\ldots(73)$$

Example.—Using the above expression, find the efficiency of the diffuser nozzle in the first stage of a two-stage air ejector, which is supplied with steam at a gauge pressure of 100 lb. per square inch, and maintains a vacuum of 28·75 in. Hg, with a barometer of 30 in. Hg. The temperature of the air and vapour entering the suction is 79° F . and the weight of air passing is 80 lb. per hour. The steam used is 100 lb. per hour, and the intermediate vacuum between the stages of the ejector is 25·5 in. Hg.

First, find the mass of vapour which is associated with the air at the ejector suction.

The partial air pressure p_a is

$$p_a = (\text{vapour pressure corresponding to } 79^\circ \text{ F.}) - 28{\cdot}75$$

$$= 29 - 28{\cdot}75$$

$$= 0{\cdot}25 \text{ in. Hg.}$$

$$\text{Then} \quad V_a = (12{\cdot}39 \times 30)(459{\cdot}6 + 79)/[0{\cdot}25(459{\cdot}6 \times 32)]$$

$$= 1628.$$

From the steam tables the volume per pound of vapour at 79° F. = 657 c. ft.

The weight of vapour therefore associated with 1 lb. of air will be

$$= 1628/657$$

$$= 2{\cdot}48 \text{ lb. per pound.}$$

The total weight of air and vapour entering the ejector suction will be

$$= (1 + 2{\cdot}48)80$$

$$= 278{\cdot}4 \text{ lb. per hour.}$$

Then w, the weight of air and vapour per pound of steam, will be

$$= 278{\cdot}4/100$$

$$= 2{\cdot}784 \text{ lb. per pound.}$$

Taking $n = 1\cdot3$, $E = 0\cdot85$, $P = 114\cdot7$ lb. per square inch.,

$$\begin{aligned} \text{and}\quad P_1 &= (30 - 28\cdot75) \times 14\cdot7/30 \\ &= 0\cdot613 \text{ lb. per square inch,} \\ \text{and}\quad P_d &= (30 - 25\cdot5) \times 14\cdot7/30 \\ &= 2\cdot205 \text{ lb. per square inch,} \end{aligned}$$

we have, by substituting in equation (73),

$$\begin{aligned} 2\cdot79 &= 0\cdot85E_d \\ &\quad \times[\{114\cdot7^{(1\cdot3-1)/1\cdot3} - 0\cdot613^{(1\cdot3-1)/1\cdot3}\}/\{2\cdot205^{(1\cdot3-1)/1\cdot3} - 0\cdot613^{(1\cdot3-1)/1\cdot3}\}] - 1 \\ &= 0\cdot85E_d[(2\cdot987 - 0\cdot893)/(1\cdot2 - 0\cdot893)] - 1 \\ &= 0\cdot85E_d \times 6\cdot82 - 1, \\ \text{or}\quad E_d &= 3\cdot79/(0\cdot85 \times 6\cdot82) \\ &= 0\cdot65 \text{ or } 65 \text{ per cent.} \end{aligned}$$

Example.—Find the weight of air and vapour which is dealt with by a single-stage air ejector of the reversed nozzle type, when the ejector is maintaining a vacuum of 26 in. Hg at a suction temperature of 100° F. The steam used by the ejector is 550 lb. per hour, and the supply steam pressure is 140 lb. per square inch gauge. The ejector discharges against a pressure of 16 lb. per square inch, and the efficiencies of the steam nozzle and diffuser nozzle are 85 per cent and 70 per cent respectively.

From equation (73) the weight of air and vapour w is:

$$\begin{aligned} w &= EE_d[\{P^{(n-1)/n} - P_1^{(n-1)/n}\}/\{P_d^{(n-1)/n} - P_1^{(n-1)/n}\}] - 1 \\ &= 0\cdot85 \times 0\cdot7[154\cdot7^{(1\cdot3-1)/1\cdot3} - \{(30 - 26)14\cdot7/30\}^{(1\cdot3-1)/1\cdot3}] \\ &\qquad \div [16^{(1\cdot3-1)/1\cdot3} - \{(30 - 26)14\cdot7/30\}^{(1\cdot3-1)/1\cdot3}] - 1 \\ &= 0\cdot85 \times 0\cdot7[(3\cdot2 - 1\cdot168)/(1\cdot892 - 1\cdot168)] - 1 \\ &= 2\cdot28 - 1 \\ &= 1\cdot28 \text{ lb. per pound of steam.} \end{aligned}$$

The total weight of air and vapour dealt with will be

$$\begin{aligned} &1\cdot28 \times 550 \\ &= 704 \text{ lb. per hour.} \end{aligned}$$

The partial air pressure p_a is

$$\begin{aligned} p_a &= 28\cdot07 - 26 \\ &= 2\cdot07 \text{ in. Hg.} \\ \text{Then}\qquad V_a &= (12\cdot39 \times 30)(459\cdot6 + 100)/[2\cdot07(459\cdot6 + 32)] \\ &= 204\cdot4 \text{ c. ft. per pound.} \end{aligned}$$

The specific volume of the vapour at 100° F. is 350·8 c. ft. per pound, and the weight of associated vapour per pound of air will therefore be

$$\begin{aligned} &= 204\cdot4/350\cdot8 \\ &= 0\cdot5825 \text{ lb. per pound.} \end{aligned}$$

The total weight of dry air dealt with will be

$$= 704/(1 + 0{\cdot}5825)$$
$$= 445 \text{ lb. per hour,}$$

and the total weight of vapour will be

$$704 - 445$$
$$= 259 \text{ lb. per hour.}$$

MIRRLEES AIR EJECTORS

Fig. 53 illustrates the simple type of air ejector as manufactured by Messrs. The Mirrlees Watson Company, Glasgow, and known as Mirrlees-Le Blanc Multi-jector Air Pump. The simple type is so named because no inter-condenser is employed between the stages to condense the steam used in the first stage.

F is the suction branch which is connected to the condenser, C the first stage steam nozzle, and D the first stage compression pipe or diffuser. The steam supply to the first stage nozzle C is through the steam pipe H and strainer A. The steam issuing from the nozzle C entrains the air and vapour entering the suction branch F, the whole mass of air, vapour, and steam is discharged into the compression pipe with considerable velocity and a small pressure rise is obtained in the diffuser E. There is, however, a considerable residual velocity which reduces the momentum imparted by the second stage nozzles. These nozzles are shown at C_2; B_2 is the second stage suction chamber, D_2 the second stage compression pipe, and E_2 the second stage diffuser nozzle. G is the main steam supply to the second stage steam chest A_2, and from this steam chest the first stage steam supply is taken. The steam issuing from the second stage steam nozzles

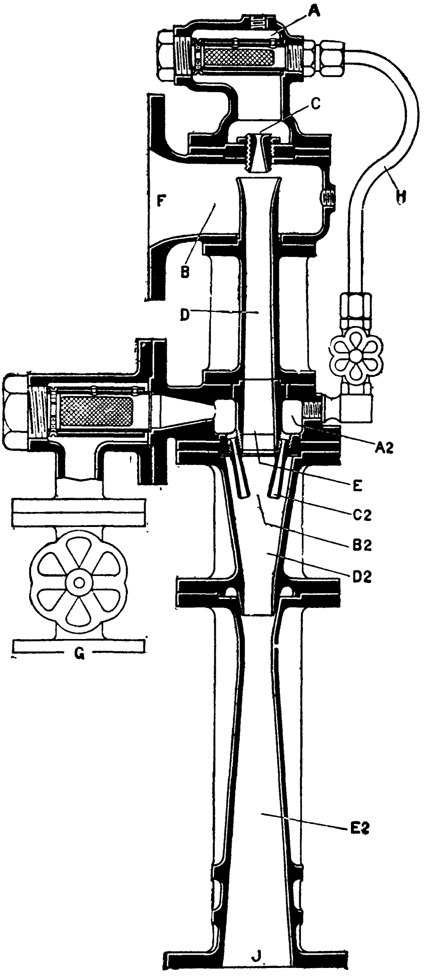


Fig. 53.—Mirrlees-Le Blanc Multi-jector Air Pump —Simple Type

C_2 entrains the total mass of air, vapour, and steam discharged from the first stage, and compresses this mass along with its own steam in the diffuser E_2, where the kinetic energy is transformed into useful work in overcoming the atmospheric pressure at the discharge end J. This arrangement gives a very simple piece of apparatus, merely requiring three connections, these being the air suction pipe to the condenser, the steam supply pipe, and the discharge pipe. To start this ejector the makers recommend that the stop valve G be first opened and then the valve on the air suction branch F. The vacuum will then quickly rise and become stationary. When this is observed the steam supply valve to the first stage should be opened and the maximum vacuum, consistent with the air leakage and the temperature conditions prevailing on the plant, will then be obtained. These standard multi-ejectors are designed and manufactured for a minimum operating steam pressure of 85 lb. per square inch gauge.

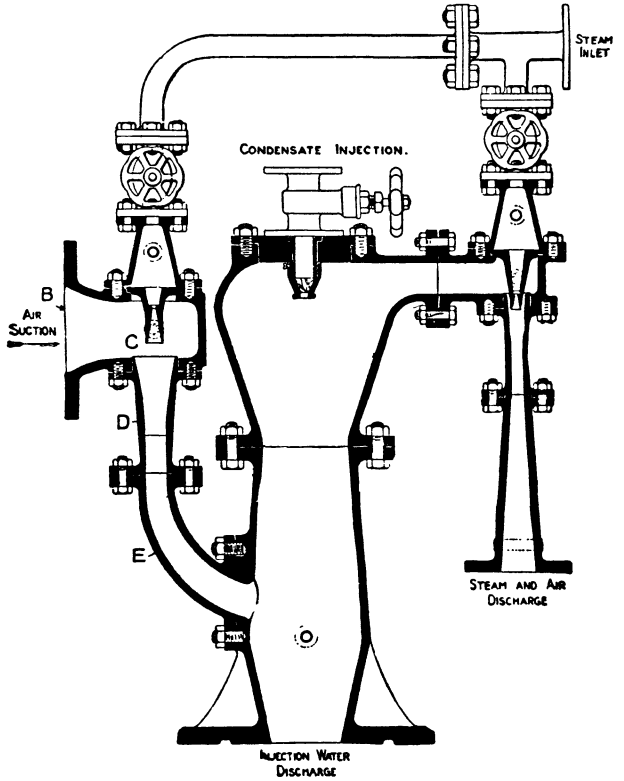


Fig. 54.—Mirrlees Inter-condenser Type of Air Ejector

Fig. 54 illustrates the central-condenser or inter-condenser type of air ejector as manufactured by Messrs. The Mirrlees Watson Company. This type of ejector is more economical in steam than the simple type described, owing to the steam used in the first stage being condensed in the central condenser and not having to be ejected by the second stage. Two stages are employed with a condenser situated between the stages. B is the air suction which is connected to the main condenser, C is a steam nozzle, D a convergent compression pipe, and E a diffuser nozzle. The air and vapour entering the air suction B is entrained by the steam issuing from the steam nozzle C, and discharged through the compression pipe D and diffuser E to the bottom of the central condenser. At the top of the central condenser there is a spraying box which receives condensate usually from the discharge of the water-extraction pump. This condensate is sprayed in the spraying box and falling down through the condenser, condenses the steam discharged from the diffuser E of the first stage, and cools the rising air, thus increasing its density. This cooled air and its associated vapour pass to the second stage, where a second compression takes place and the air is discharged to the atmosphere. The injection water and condensed steam from the first stage pass out at the bottom of central condenser to the water-extraction pump suction pipe, through a looped pipe which is vented to the main condenser by a small vent pipe. The vacuum carried in the central condenser is normally 24 to 25 in. Hg, which gives a compression of about 5 to 1, depending upon the vacuum being carried in the main condenser.

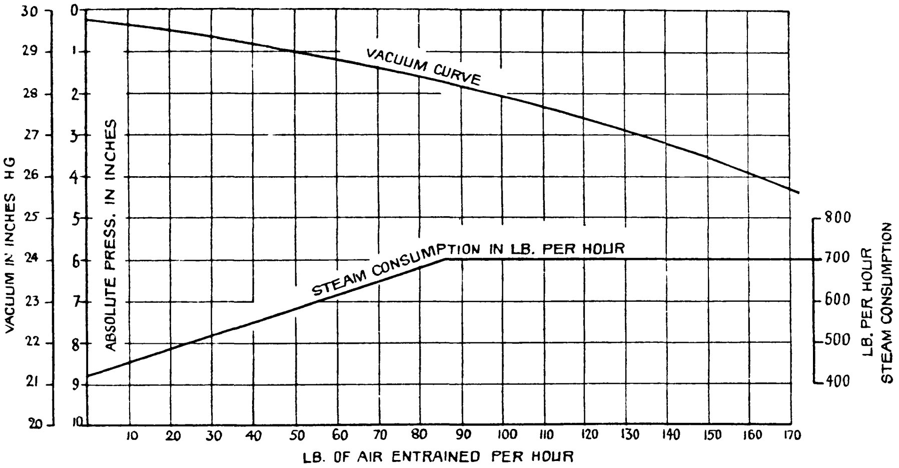


Fig. 55.—Mirrlees Test Chart

The steam chests, nozzle holders, and nozzles are made of a special hard close-grained gun-metal suitable for high pressure steam and superheat. The convergent compression pipes and water spraying box are of gun-metal. The remaining parts of the ejector are of close-grained cast iron.

When starting up this type of ejector a supplementary supply of cold water is necessary for the central condenser. Once the main unit is running and the water-extraction pump is receiving condensate from the main condenser this supplementary supply can be shut off, and the condensation in the central condenser effected by the condensate discharged from the water-extraction pump.

Messrs. The Mirrlees Watson Company manufacture two styles of this type of air ejector. One, which is styled type A compression, uses 17 per cent of the operating steam in the first stage and the remaining 83 per cent in the second stage; the other, which is styled type B compression, uses approximately 50 per cent of the operating steam in each stage at full load. Type B compression has the great advantage that at reduced air leakages the steam supply to the first stage ejector can be cut down, thereby effecting an economy in steam and giving a certain degree of flexibility to the apparatus.

Fig. 55 shows a standard test chart from a central-condenser type of air ejector as manufactured by Messrs. The Mirrlees Watson Company, and having a maximum steam consumption of 700 lb. per hour. This chart clearly indicates how a great saving in steam can be effected by cutting down the steam supply to the first stage when the air being handled falls below 85 lb. per hour.

An air ejector of the simple type and having a performance equal to that shown in fig. 55 would have a steam consumption of approximately twice that given above for the central-condenser type.

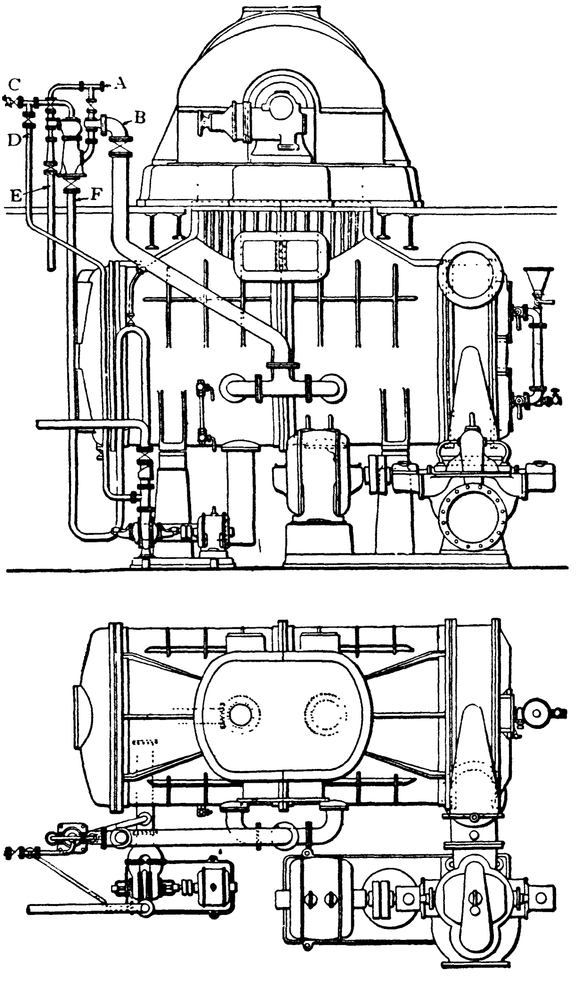


Fig. 56.—Mirrlees Surface Condenser, with Central Condenser Type of Air Ejector

APPLICATION TO SURFACE CONDENSERS

Fig. 56 illustrates the application of the central-condenser type of air ejector to a surface condenser.

The ejector is carried on the turbine floor-level, so that the operating valves are close to the turbine attendant, and the central condenser is easily drained to the water-extraction pump suction piping. A is the steam supply to the ejectors; B the air suction piping from the main condenser; C the supplementary water supply for starting up purposes; D the condensate discharge from the water-extraction pump to the central condenser; E the discharge from the second stage ejector; and F is the central condenser drain to the water-extraction pump suction. It will be noted that this drain pipe is looped so as to form a water seal. The reason for this is because of the difference in vacuum between that in the main condenser and that in the central condenser; without this seal or loop air, which had already been extracted and compressed in the first stage ejector, would return to the main condenser only to be extracted again by the first stage ejector. An internal circulation of air would thus be set up which would reduce the effective air handling capacity of the ejector. The height of this loop should exceed the equivalent head of water of the difference in vacuum between that in the main condenser and that in the central condenser.

The drainage from the central condenser is at a temperature greater than the vacuum temperature in the main condenser; a percentage of the drainage water will therefore pass into vapour, as the drainage water comes under reduced pressure, in passing through the loop or seal. This vapour must be released, and if the main condenser is fitted with an integral air cooler the seal should be released to the main condenser. If no air cooler be fitted to the main condenser it may be released to the ejector suction pipe as shown, otherwise the effectiveness of the intergral air cooler would be nullified by the injection of this relative hot vapour into the cold ejector suction pipe.

HICK-BREGUET EJECTORS

The air ejectors made by Messrs. Hick Hargreaves, Limited, Bolton, are manufactured under the patents of M. Delaporte of Messrs. Breguet, Paris. This ejector is of the two-stage type with an intermediate jet condenser to condense the steam used in the first stage, and is very similar to the Mirrlees central-condenser type of air ejector previously described. The main difference lies in the automatic spring loaded air inlet valve situated between the stages. This valve admits air when the vacuum in the intermediate condenser rises above 26 in. Hg, thus maintaining the compression ratio in the second stage ejector under 7·5 (see p. 131) to ensure stability of action.

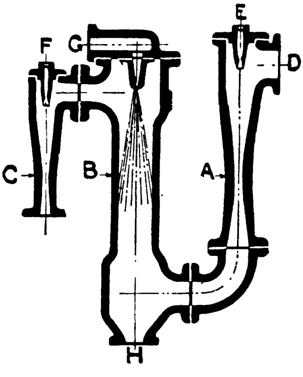


Fig. 57.—Section of Hick-Breguet Ejector

The ejector is shown in section in fig. 57 and in outside view in fig. 58. A is the first stage ejector; C is the second stage ejector; B is the intermediate jet condenser; D is the suction branch which is connected to the condenser; E is the steam supply to the first stage nozzle; F is the steam supply to the second stage nozzle; G is the water inlet to the intermediate condenser; and H is the drain from the intermediate condenser to the main

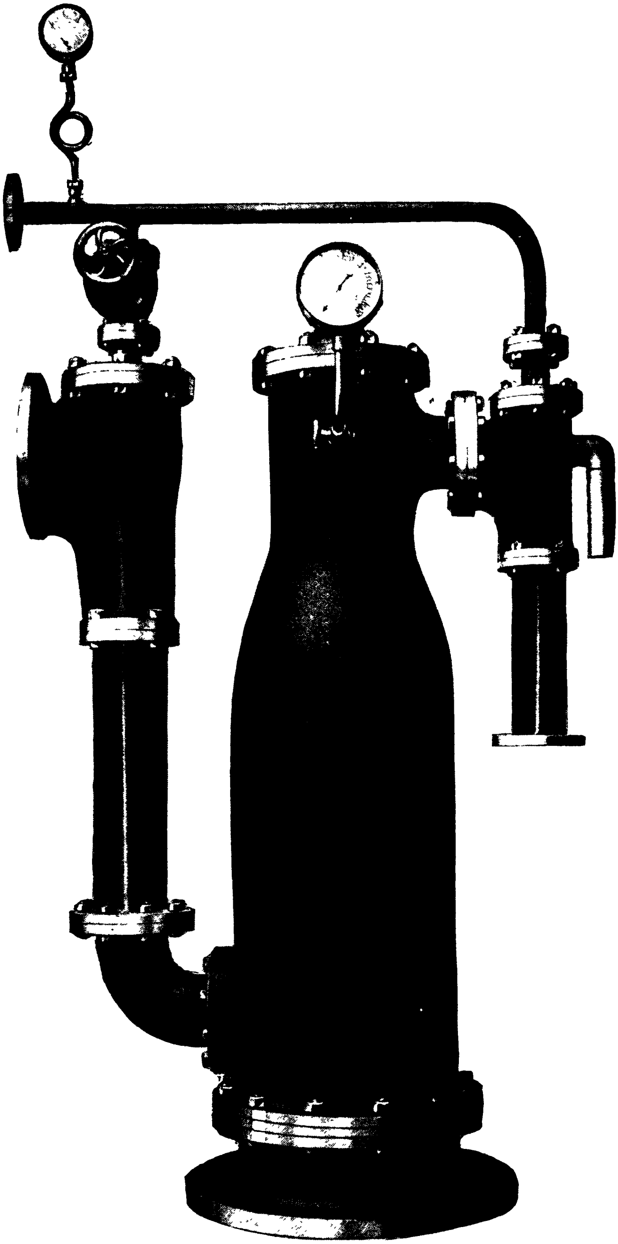


Fig. 58.—Hick-Breguet Ejector

condenser. The automatic air inlet is shown in fig. 58 at the side of the second stage ejector. This air inlet has a spring controlled valve which

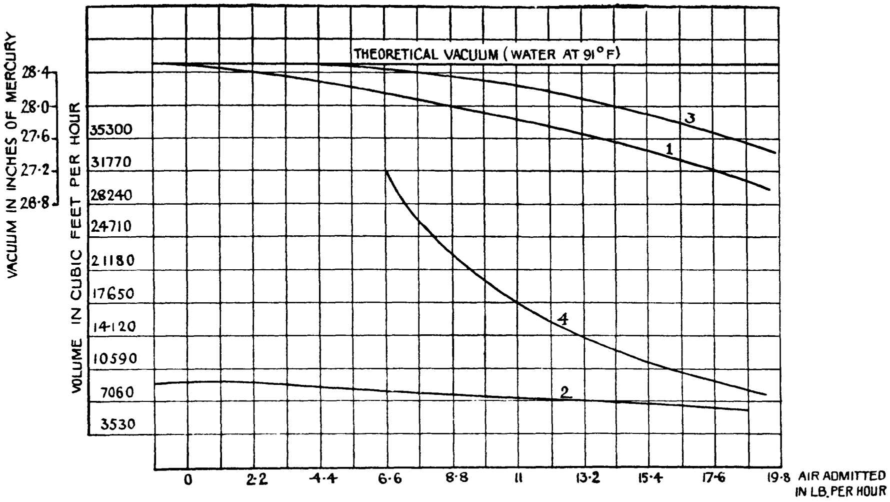


Fig. 59.—Hick-Breguet Tests

Curve No. 1 shows Vacuum obtained with Water leaving the Condenser at 91° F. Curve No. 2 gives the corresponding Volumes. Curve No. 3 shows Vacuum obtained with Water leaving the Condenser at 91° F., the Cooler being fed with Water at 64 5° F. Curve No. 4 gives the corresponding Volumes.

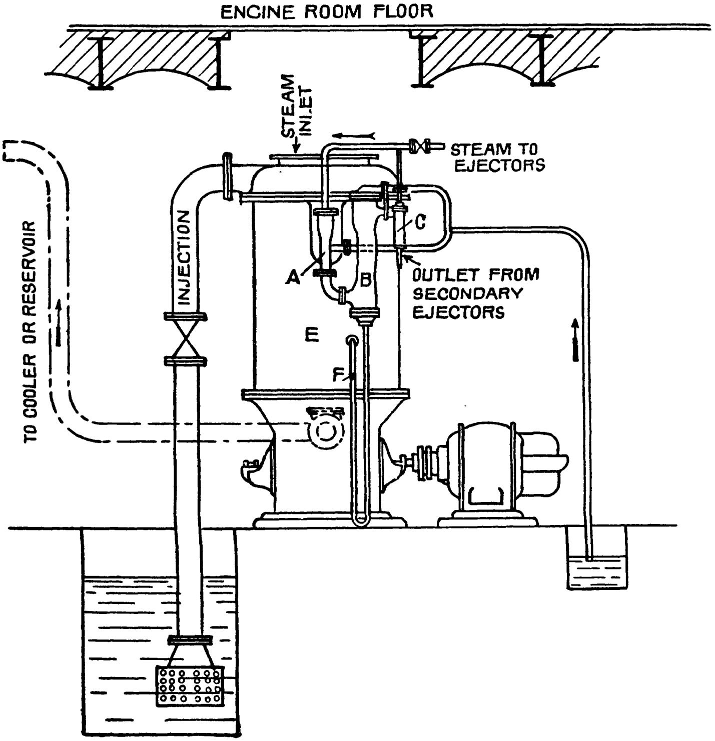


Fig. 60.—Arrangement of Jet Condenser, fitted with Breguet Ejector

has an adjustment so that it may be set to open at any desired vacuum. These ejectors are normally designed with a steam consumption ratio between the second and first stages of 2 to 1, that is, the second stage takes two-thirds of the total steam used by the ejector. The makers claim that these ejectors will work ordinarily at any pressure above 55 lb. per square inch, but where the pressure is below this limit a specially designed ejector must be used.

Fig. 59 shows the results obtained on test by an ejector using 193 lb. of steam per hour when working in conjunction with a jet condenser dealing with 22,000 gall. of injection water per hour. Known quantities of air were admitted through specially calibrated pipes and the vacua recorded with these air leakages were noted.

These curves are self explanatory and show the air handling capacity of this apparatus.

Fig. 60 shows the application of the ejector to a Hick-Breguet jet condenser. A is the first stage ejector; C is the second stage ejector; B is the intermediate jet condenser; and E is the main jet condenser. The water for the intermediate condenser is taken from main condenser injection. The loop pipe or seal through which the intermediate condenser drains to the main condenser is shown at F.

WEIR AIR EJECTORS

Messrs. G. & J. Weir, Ltd., Glasgow, manufacture both the two-stage simple and the two-stage with inter-condenser types of air ejectors. The standard ejectors manufactured by this firm are designed for a minimum steam pressure of 150 lb. per square inch gauge.

Fig. 61 shows the standard simple type. Two stages are employed with a steam consumption ratio of 1 to 20 between the first and second stages.

A is the air suction which is connected to the main condenser; B is the first stage steam nozzle; C is the second stage steam nozzle; D is the first stage diffuser; E is the second stage diffuser; F is the first stage air suction chamber which incorporates a light non-return valve G. This valve comes into action when the steam pressure falls below the designed pressure and prevents a sudden fall in the vacuum. When the steam pressure is 150 lb. per square inch or over, the valve is open and inoperative. H is the main steam supply, the first stage steam nozzle being fed through the pipe J. The steam nozzles are arranged as a group of very small convergent-divergent nozzles to present a large jet surface to the air and vapour. The entrainment of the air and vapour is accomplished by the friction of the high velocity jet of steam issuing from the steam nozzles. There are practical limitations, however, which limit the size to which these nozzles can be reduced quite independent of a decrease in nozzle efficiency with decrease in size. The liability of very small nozzles to become choked with dirt or scale off the steam pipes limits their size, and to counteract this evil it is a customary practice to fit a strainer in the steam pipe to strain the steam before it passes to the steam nozzles. These small nozzles, as manufactured by Messrs. G. & J. Weir, Ltd., are made with a throat diameter of 3/32 in. and the holes in the steam strainer 1/16 in. in diameter. This combination has been found to give no trouble in practical operation. The material of which these nozzles are made is Monel metal, which is a copper nickel natural alloy, having a composition of 27–29 per cent copper, 68–70 per cent nickel, and 4–5 per cent manganese. This metal is non-corrosive and is highly resistant to the erosion caused by high velocity steam. The diffuser nozzles are made in hard gun-metal. When the ejector has to operate with superheated steam, all parts which come into contact with the steam are made in steel, except the steam valve faces which are of Monel metal.

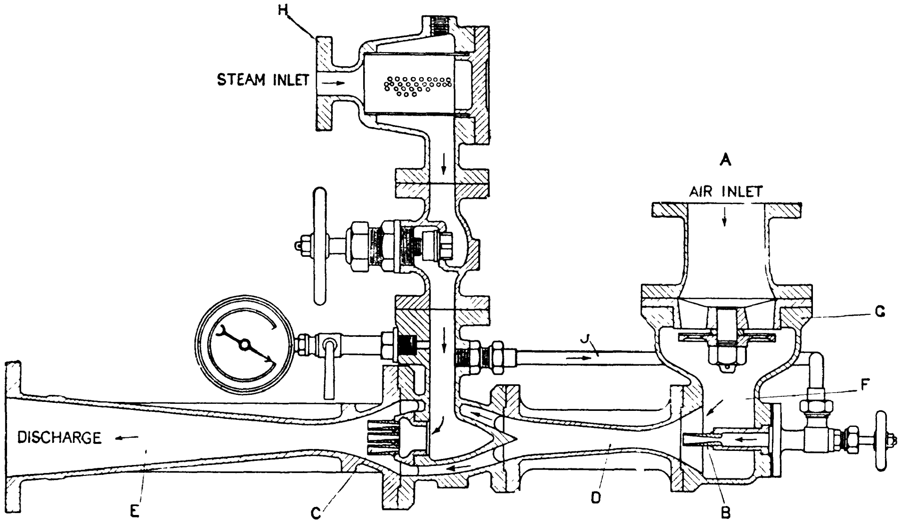


Fig. 61.—Weir Standard Simple Type of Air Ejector

Fig. 62 shows the inter-condenser type of air ejector as made by Messrs. G. & J. Weir, Ltd. This figure is self explanatory, and the first and second stage ejectors follow generally the description given for the simple type. The inter-condenser is of the surface type having $\frac{3}{4}$-in. outside diameter condenser tubes, ferruled into brass tube plates to allow for expansion and contraction, and is similar to any small ordinary condenser. The cooling water for the condenser is taken from the cooling water system of the main condenser for marine installations, and from the condensate pump discharge for land installations and enters at the bottom as shown; it makes four passes and leaves at the top. The steam, air, and vapour discharged from the first stage enters at the top right-hand end, and the cooled air with its associated vapour and the condensed steam leave at the bottom left-hand end. The outlet pipe forms a loop, the air and vapour passing upwards to the second

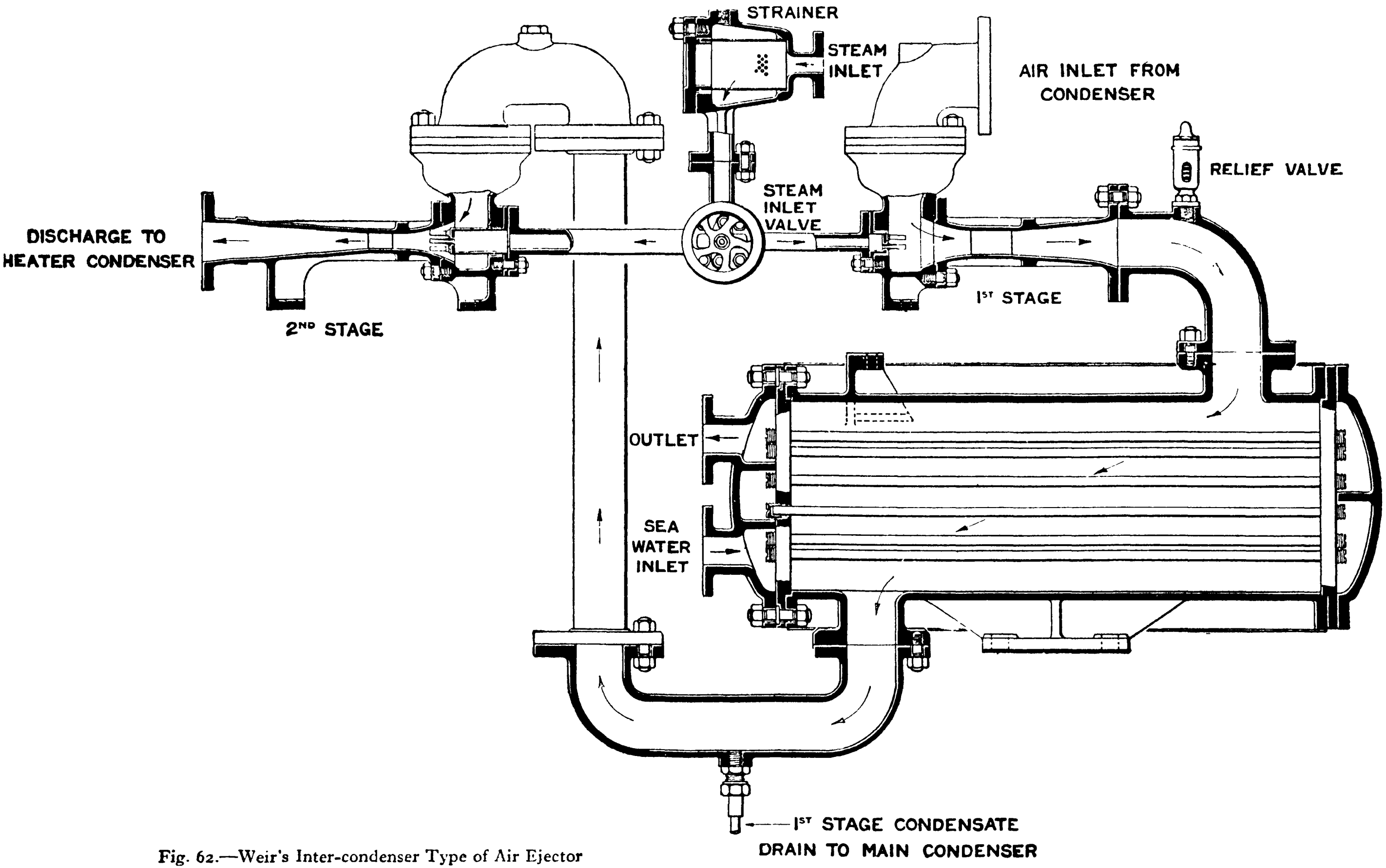


Fig. 62.—Weir's Inter-condenser Type of Air Ejector

stage inlet, while the condensed steam is drained to the main condenser by a small bore drain pipe. No attempt is made by the makers to loop this drain pipe in a manner similar to that adopted for the Mirrlees Watson multi-jectors or for the Hick-Breguet ejectors, because of the very small bore of the pipe. The bore is so chosen that the pipe runs practically full, and whatever leak back of air there may be to the main condenser has been found from practical experience to have no detrimental effect on the vacuum. This type of air ejector is specially suited for marine conditions, because of the adoption of surface inter-condensers with a circulation of sea water.

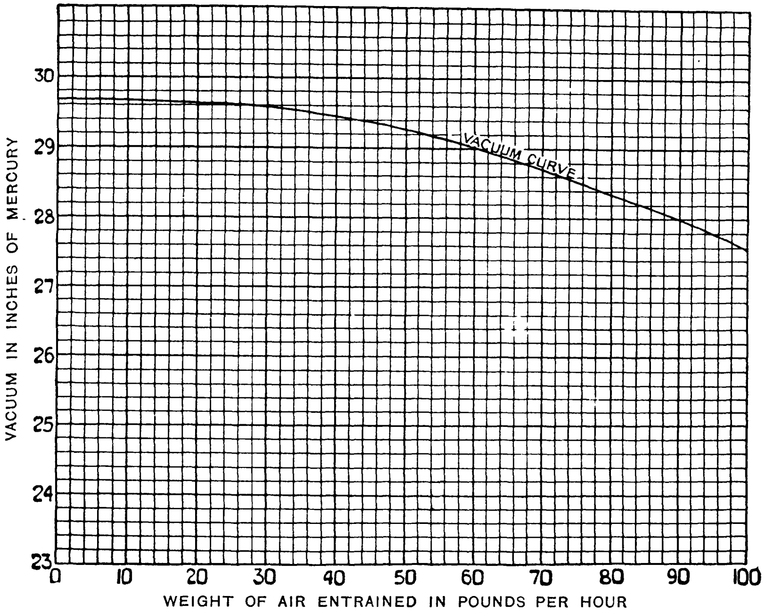


Fig. 63. Air Capacity Test Sheet to Air Ejector

One of the duties demanded of an air ejector fitted to a main condenser on board ship is that during manœuvring or during a sudden shut down of the main engines the air ejector must maintain the vacuum. If the air ejector, therefore, be supplied with condensate as the cooling medium for its inter-condenser, then when the main engines are stopped the cooling water supply to the inter-condenser is stopped, or so far reduced as to upset the working of the ejector, and thus materially impair the vacuum. A supplementary cooling water supply would need to be provided, which means added complication. The heat in the first stage ejector discharge is, however, lost with Messrs. G. & J. Weir's design of inter-condenser as fitted on board ship, and no claim is made by the makers for its recovery; the loop pipe arrangement with release to the main condenser is equally bad in this respect, because of the self evaporation of the drain water from the jet intermediate condenser, owing to the reduction in pressure which the drain undergoes in flowing from the intermediate condenser to the top of the loop.

The Weir air ejectors are started by simply opening the steam valve.

The steam consumption ratio of a Weir air ejector of the inter-condenser design between the first and second stages is about 1 to 2·3, and a works test sheet showing the air handling capacity is shown in fig. 63. The ejector under test had a steam consumption of 560 lb. per hour, the steam supply pressure being 150 lb. per square inch gauge.

RADOJET AIR EJECTORS

The Radojet air ejector is of American origin, and is handled in this country by Messrs. John Musgrave & Sons, Limited, Bolton. The Radojet ejector or air pump takes its name from the form of steam jet and diffuser employed; the flow is radial, the steam being projected by the steam nozzle in a thin sheet or disc into the radial flow diffuser nozzle. The radial flow type of ejector is only used for the second stage, the first stage being a normal tubular reversed nozzle ejector. They are manufactured with or without inter-condensers, and suitable for steam pressures from 80 lb. per square inch upwards.

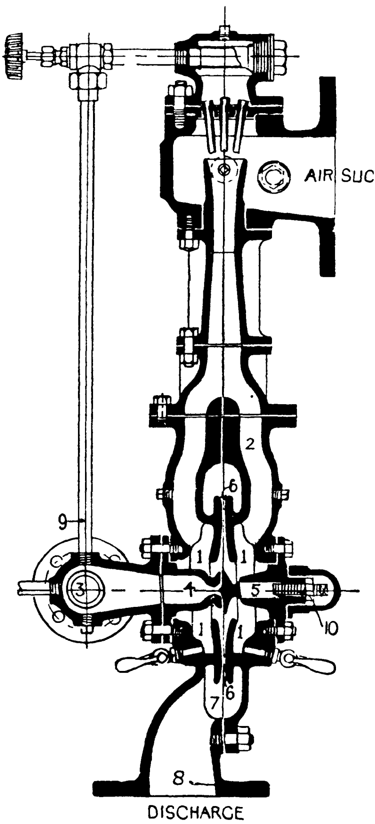


Fig. 64—Two-stage Radojet Air Ejector

Fig. 64 shows a two-stage Radojet without inter-condenser. The first stage is at the top, and it will be seen that this is a normal tubular ejector with a group of small convergent-divergent steam nozzles. The discharge from the first stage passes to the second stage suction chamber 1 by the ducts 2. The steam enters at 3, and enters the second stage steam nozzle 4; the throat of this steam nozzle is circular and of constant area, the divergent part of the steam nozzle is, however, disc shaped by the flow taking place between the disc 5 and the disc-shaped end of the steam nozzle 4. The diffuser 6 is similarly formed by the disc-shaped diffuser walls. The steam leaves the steam nozzle as a high velocity disc of steam with radial flow, and passing across the suction chambers 1 entrains the air and vapour by surface friction on both sides of this disc of high velocity steam. The combined mass of air, vapour, and steam enters the radial flow diffuser where the kinetic energy is transformed into pressure energy by the gradually increasing area of the diffuser. The radial diffuser discharges directly into the exhaust belt 7, and thence to the discharge 8. The first stage ejector is fed from the steam inlet 3 by the small vertical pipe 9 passing to the first stage nozzles. One of the characteristics of this ejector is the variable expansion which the construction makes possible in the radial steam nozzle.

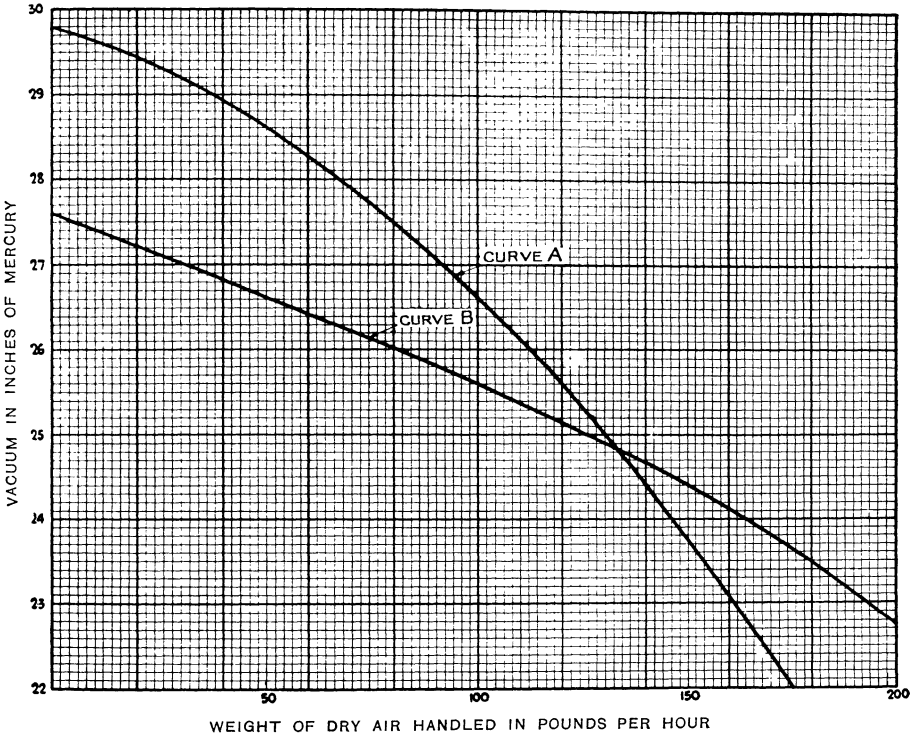


Fig. 65.—Dry Air Handling Capacity of a Two-stage Radojet Air Ejector

It will be noticed that the right-hand half, 5, of the radial steam nozzle is screwed at 10 into the casing, so that by rotating this half of the nozzle by the square cut on the spindle end of the nozzle the discs forming the diverging part of the steam nozzle can be brought closer together or farther apart as desired. Since the throat area is constant this adjustment provides a steam nozzle with a variable or adjustable expansion ratio.

Fig. 65 shows the dry air handling capacity of such a two-stage Radojet without an inter-condenser, when supplied with dry saturated steam. Curve A shows the performance of a two-stage Radojet, and curve B shows the performance of a single-stage Radojet having the same steam consumption as the two-stage apparatus. These curves are of interest since they intersect at a vacuum of about 25 in. Hg, thus showing that for vacua of 25 in. Hg and under only a single stage should be employed.

The Radojet air pump is manufactured with an inter-condenser of both the surface and jet types.

Fig. 66 shows the design adopted for the surface type of inter-condenser air pump.

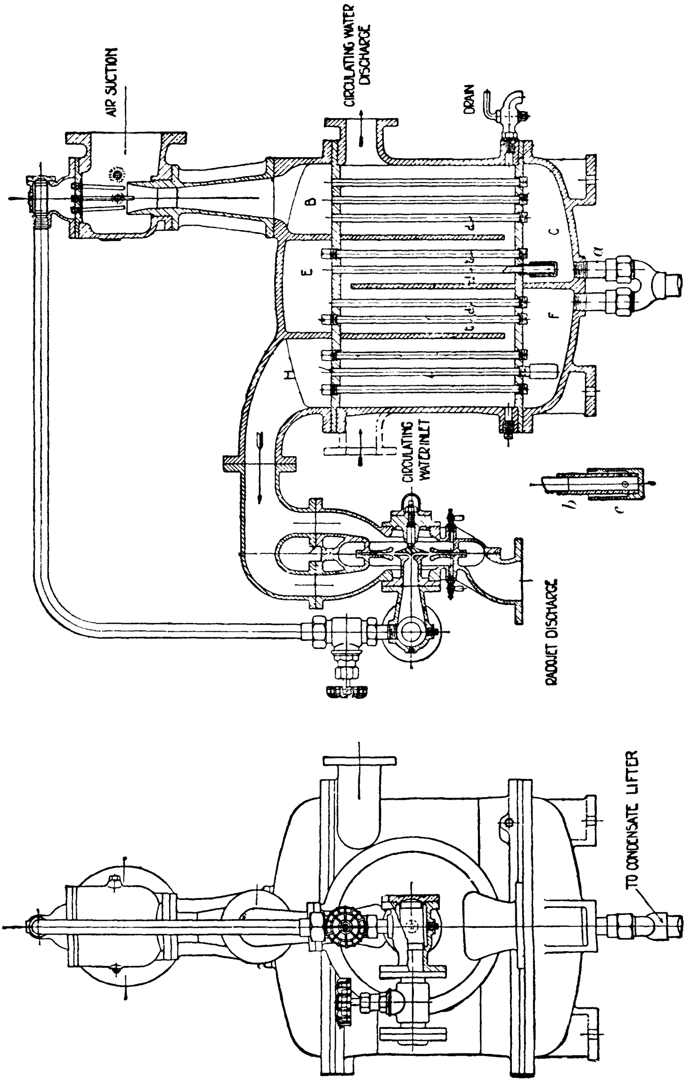


Fig. 66.—Musgrave's Radojet Air Ejector

The first and second stages follow the general lines adopted for their air pumps without an inter-condenser, as already described. The surface inter-condenser is of the tubular type with the tubes placed vertically. These tubes are expanded into the top tube plate, and ferruled into the bottom tube plate to allow for expansion. The steam is inside the tubes, and the cooling water is outside the tubes. The illustration shows a four-pass condenser; the number of passes adopted, however, varies from two to six, depending upon the size and operating conditions of the air pump.

Referring to fig. 66 the operation is as follows. The discharge from the first stage ejector enters the compartment B and flows downwards through the inside of the tubes *d*, where the greater proportion of the steam is condensed. The condensed steam drains to the bottom of the compartment C, and is drained away by the drain pipe *a*, the remaining portion of the steam along with the air passes upwards through the tubes *t*, where a further portion of the steam is condensed and drains back to the compartment C by the drain tubes *d* of the second pass. These drain tubes are fitted at the bottom with a water seal, as shown at *b* and *c*, so that no upward flow can take place in the drain tubes *d* because of the higher pressure carried in the compartment C. The process is repeated in the two following passes, the air and its associated vapour emerging into the compartment H in a cooled condition, while the condensed steam is drained from the compartment F. The circulating water is arranged contra-flow to the air, steam, and vapour, to give the maximum degree of cooling to the air. The air and vapour then pass to the suction of the second stage Radojet.

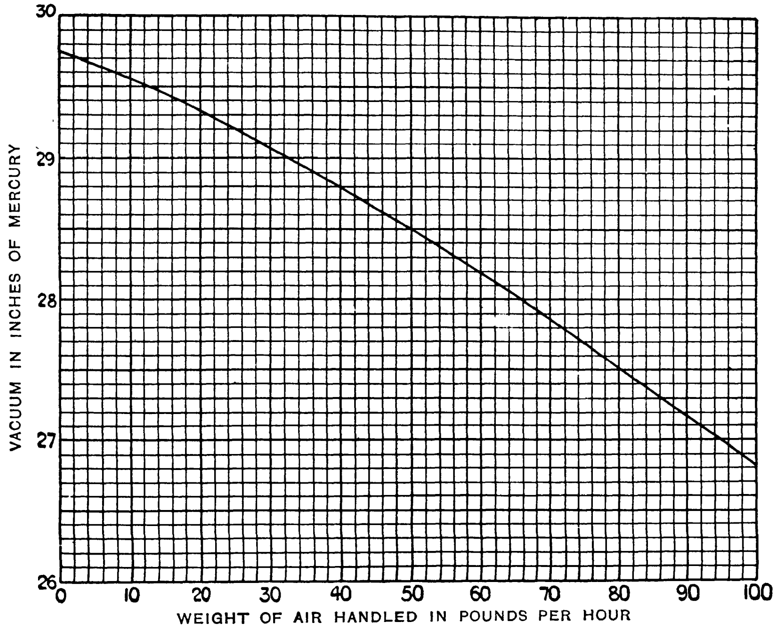


Fig. 67.—Dry Air Handling of a Two-stage Radojet with Surface Inter-condenser

Fig. 67 shows the dry air handling performance of a two-stage Radojet with surface inter-condenser when supplied with dry saturated steam at 125 lb. per square inch gauge, and exhausting against a back pressure of ½ lb. per square inch gauge. The relative steam consumptions of the Radojet air pump with inter-condenser, and without an inter-

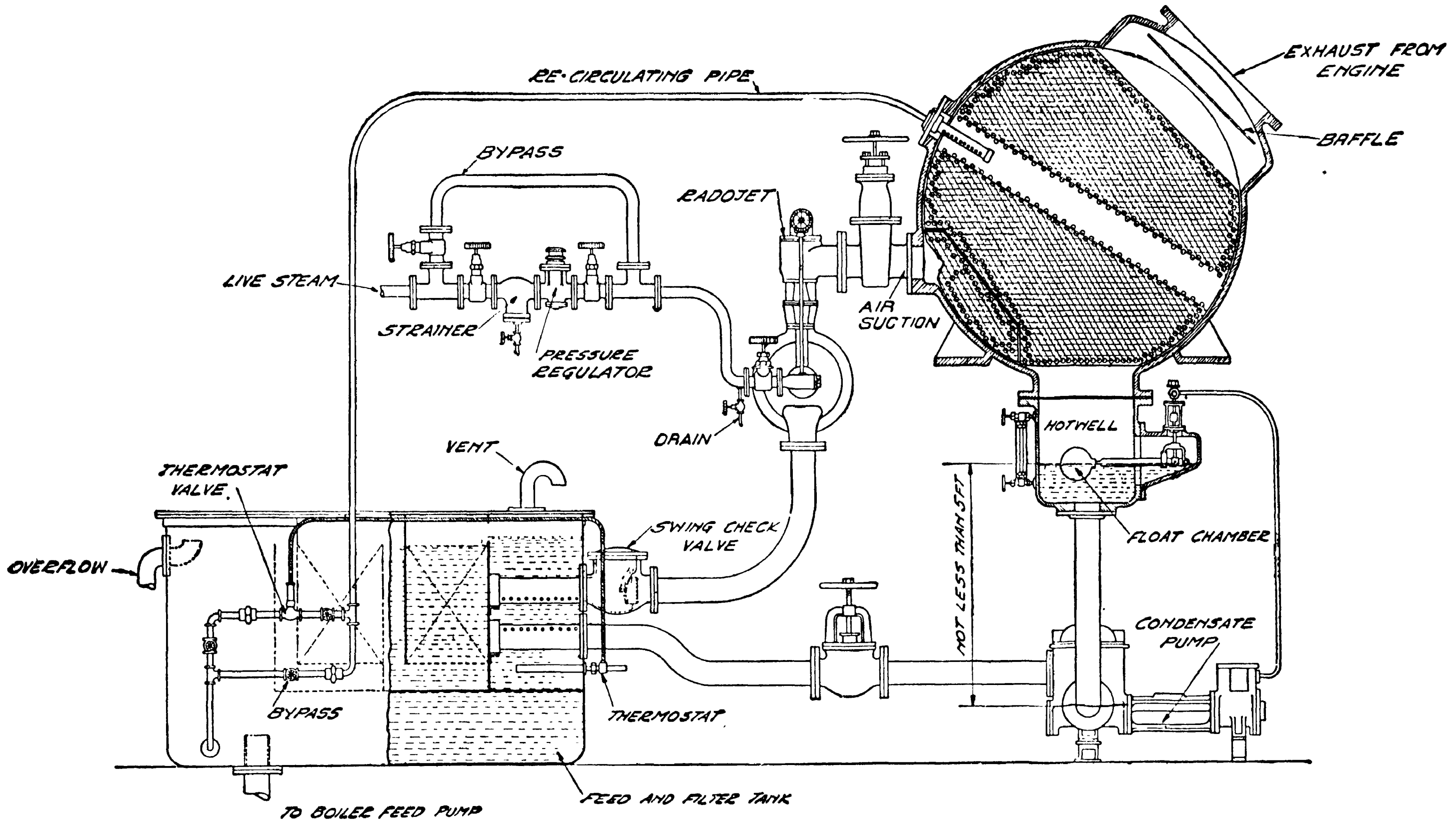


Fig. 68.—Method of Installing the Radojet Air Pump to a Surface Condenser

condenser, is approximately one-half for the same air handling capacity.

When starting up the Radojet the makers recommend that the second stage should be put into operation first, and when the vacuum reaches 25 in. Hg for the Radojet without inter-condenser, and 20 in. Hg for the Radojet with inter-condenser, the first stage should be put into operation.

Fig. 68 shows the method of installing the Radojet air pump to a surface condenser.

The supply steam to the Radojet is passed through a pressure regulator, so that a constant steam pressure is supplied to the nozzles. A gate valve is fitted between the Radojet suction and the condenser, which should be opened before starting up the air pump. The discharge is led through a swing check valve to the feed tank, and the end of the discharge pipe which is inside the feed tank is perforated on its under side. The condensate pump discharge pipe enters the feed tank immediately under the Radojet discharge pipe, and the internal pipe is perforated on its top side. When the Radojet and condensate pump are operating the steam issuing from the perforations on the top pipe meets the cold condensate issuing from the perforations on the bottom pipe, which results in a quiet and complete condensation of the discharge steam. The air rises up through the water and escapes by the vent pipe, the water passing on to the filters. The condensate pump illustrated is of the direct-acting type, float controlled from the hotwell, and bolted to the bottom of the condenser. The small float operates the steam throttle valve which admits steam to the pump, and thus regulates the pump speed, so that there is always a head on the suction valves of about 5 ft. This head enables the pump to extract the condensate against the vacuum in the condenser. A thermostat is fitted to the feed tank, and controls by temperature the thermostat valve on the recirculating pipe. This thermostat comes into operation at low loads or when the quantity of condensate is not sufficient to condense the exhaust steam from the Radojet. When the exhaust steam cannot be condensed the thermostat heats up, which results in the thermostat valve opening, and water is drawn from the feed tank through the recirculating pipe into the main condenser; this water is sprayed down among the cold condenser tubes and thus becomes cooled, and falls into the hotwell as cold water. The condensate pump therefore receives a supply of cold water, which is discharged to the feed tank, thus condensing the exhaust steam from the Radojet. The quantity of water in circulation is automatically regulated by the thermostat, and is always just sufficient to ensure complete condensation of the exhaust steam.

DELAS AIR EXTRACTORS

All the air ejectors described so far have been of the reversed nozzle type, with approximately adiabatic compression in the diffuser nozzle. In the Delas air extractor the diffuser is water jacketed by a circulation of cold water which results in the compression approaching the isothermal, and, as was stated on p. 132, gives perfect stability by causing the throat to be virtually variable in area.

The work done with isothermal compression is less than that done with adiabatic compression, so that the work done in the diffuser will be less than that done in a diffuser of the reversed nozzle type with no water jacket. For equal air handling capacity, therefore, this ejector should use less steam than an air ejector with adiabatic compression. As an offset against this, however, the heat taken up from the steam in the water jackets is lost in the cooling water, and is therefore not available for feed heating. This quantity of lost heat is stated to be not more than 5 per cent of the total in the commercial extractor.

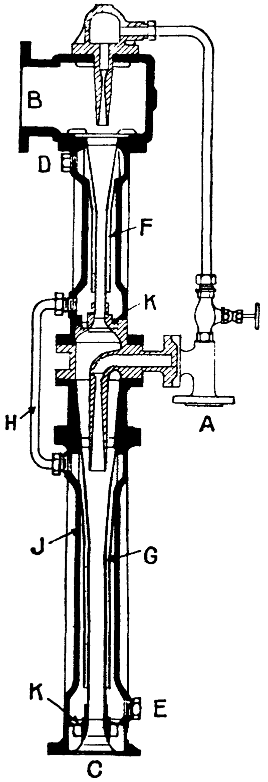


Fig. 69.—Delas Air Extractor—Simple Type

This extractor is manufactured in England under licence from Société des Condenseurs Delas by Messrs. Cole Marchant & Morley, Ltd., Bradford, and Messrs. The Brush Electrical Company, Loughborough. They are made in two types, the simple type with no inter-condenser, and the inter-condenser type.

Fig. 69 illustrates the simple type as manufactured by Messrs. Cole Marchant & Morley.

The extractor is of the two-stage type with single steam nozzles supplied from the steam inlet branch A. B is the air suction which is connected to the condenser; C is the second stage discharge to feed tank or heater. The jacket cooling water which is taken from the main circulating water inlet of the main condenser enters at D and passes down through the first stage jacket; it passes to the second stage jacket G by the pipe H, and passes down the second stage jacket and leaves at E. The diffusers are of convergent-parallel type, and are made of thin copper to allow of an easy transfer of heat through the diffuser wall.

The diffusers and jackets are encased in a cast-iron casing J to give the apparatus rigidity, and the differential longitudinal expansion between the copper diffusers and the cast-iron casing is taken up by flat diaphragm plates K.

The steam consumption ratio of the first and second stages is about 1 to 19, and the air handling capacity curve is shown in fig. 70. The steam supply pressure was 145 lb. per square inch gauge, and the steam temperature 518° F.

The latest design of two-stage extractor with intermediate condenser is shown in fig. 71. A is the air suction; B, the first stage steam nozzle; C, the second stage steam nozzle; D, the first stage diffuser; E, the second stage diffuser; F, the second stage discharge; and G, the intermediate condenser

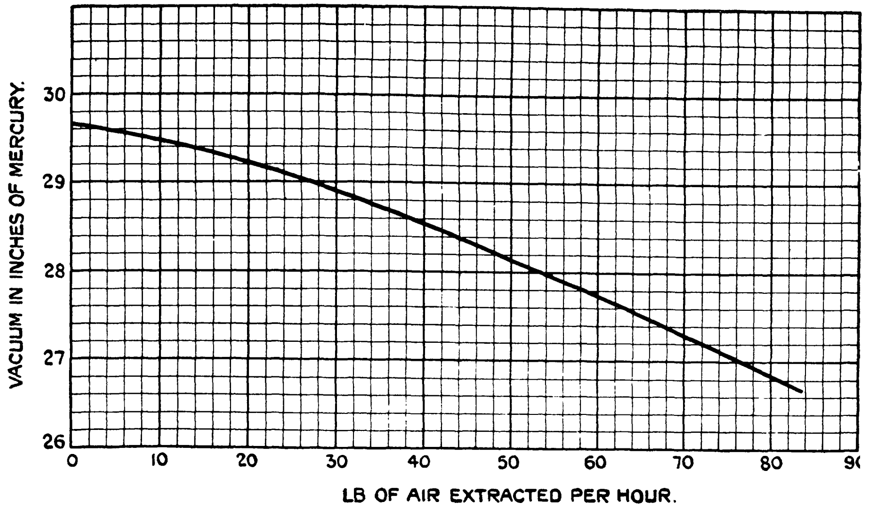


Fig. 70.—Air Handling Capacity Curve for Simple Type of Delas Air Extractor

drain to the main condenser or water-extraction pump suction pipe. The intermediate condenser is of the jet type, and is formed by a ring of holes H being drilled round the end of the first stage diffuser, so that a supply of water for condensing the steam discharged by the first stage is drawn from the first stage water jacket J. K is the second stage jacket, and in this extractor the jacket water and intermediate condenser water is condensate taken from the discharge of the water-extraction pump. The inlet for this water is at L. The water first passes upwards through the jacket K, thence it passes across by the duct M to the first stage jacket J, and flows downwards through the first-stage jacket and enters the intermediate condenser by being sprayed through the holes H. This water along with the condensed steam from the first stage falls to the drain branch G, and the air along with its associated vapour passes upwards to receive its second compression in the second stage. Both stages, along with the intermediate condenser, are encased in a single casting N, which makes for simplicity.

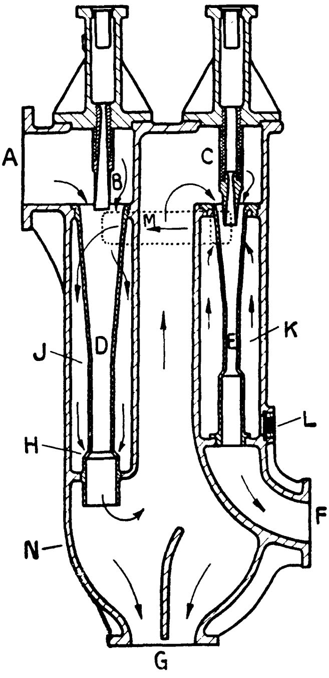


Fig. 71.—Delas Two-stage Air Extractor

Fig. 72 shows this extractor and its connections to a surface condenser. A is the extractor connected to the condenser B by the air suction pipe C; D is the steam supply branch, and E the drain to the water-extraction pump suction. This drain water first passes to a float control box F, which performs the same function as the loop pipe or U-tube previously described and illustrated in fig. 60, and allows the water to drain freely, while at the same time preventing a leak back of air to the main condenser. G is a water-extraction pump drawing the condensate from the main condenser B through the suction pipe H, and discharging through a surface heater L, which receives the discharge from the second stage ejector by the pipe M, thus conserving the heat discharged by the second stage. The circulating or cooling water for the jackets and the intermediate condenser is taken from the extraction pump discharge by the pipe N, and returns to the extraction pump suction via the drain E and float box F. When starting these extractors the cooling

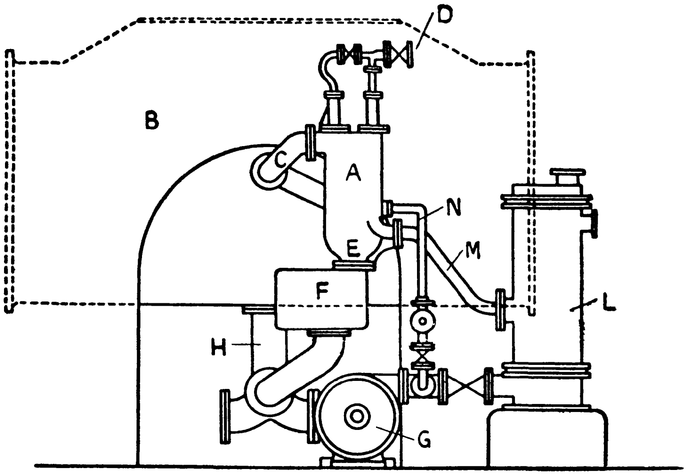


Fig. 72.—Delas Two-stage Air Extractor fitted to a Surface Condenser

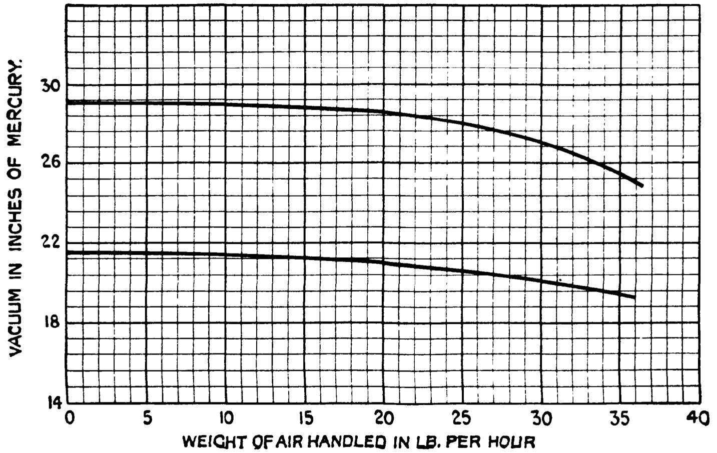


Fig. 73.—Air Handling Performance of the Cole Marchant & Morley Delas Air Extractor

water supply should be circulating before the steam is turned on.

Fig. 73 shows the air dealing performance of the latest type of Cole Marchant & Morley Delas extractor. This extractor was supplied with saturated steam at a pressure of 115 lb. per square inch gauge, and the steam consumption was 175 lb. per hour. The air being dealt with was saturated with water vapour at a temperature of 75° F. The lower curve shows the vacuum which was maintained in the intermediate condenser.

The water jacketed diffuser does not give complete isothermal compression, the actual compression line lying between the adiabatic and the isothermal. If the compression be performed along the saturation line, that is, if the temperature and pressure of the steam at any point in the compression correspond, then the numerical value of the exponent n will be $\frac{16}{15}$, and the characteristic equation for the compression will be

$$PV^{16/15} = \text{constant} = 490.$$

The formulæ given for adiabatic compression will hold for this case if the above value be given to the exponent n.

AIR AND STEAM OPERATED EJECTOR

In an arrangement of an ejector air pump adopted by Messrs. Brown, Boveri, & Co., the first stage is supplied with atmospheric air, while the second stage is supplied with high-pressure steam. The arrangement is shown in fig. 74.

A is the first stage ejector, and B is the second stage; C is the air suction branch, and D is the second stage discharge. The first stage nozzle E is fed with atmospheric air under the control of the valve F, the air entering at H. The nozzle E is of the convergent-divergent type; the air expanding through this nozzle attains a high velocity, and gives to the entrained air its first compression in the first stage diffuser. The second stage steam nozzle J is also of the convergent-divergent type and receives its steam supply at K. The second stage jet entrains the mass of entrained and operating air from the first stage, and discharges it through the second stage diffuser against atmospheric pressure. The discharge from D is led to a small condenser, where the steam is condensed and the air with its associated vapour liberated to atmosphere. This type of air pump has been fitted on the S.S. *Westphalia* with successful results.

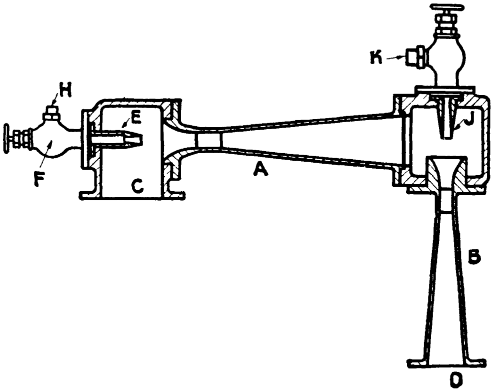


Fig. 74.—Brown-Boveri Ejector Air Pump

MIRRLEES-LE BLANC ROTARY AIR PUMP

This air pump is of the water-jet type, and has the characteristic that the necessary kinetic energy is produced in the apparatus itself. The water jet is not a solid one, but consists of a series of thin sheets or plugs of water between which are trapped the air and non-condensable gases. A section through the pump is shown in fig. 75, while fig. 42 (p. 94) shows the pump in outside view. Referring to fig. 75 the apparatus consists essentially of a rotating wheel driven by an electric-motor or other drive, and has a series of specially curved vanes arranged around the periphery of the wheel. Inside the rotating vanes is housed a distributor which has a nozzle-shaped outlet, suitably positioned so that as the rotating blades pass this nozzle they shear off slices of the water and impart to them a high velocity. These slices or thin sheets of water are discharged into a converging cone, and trap between them a certain volume of air and non-condensable gases which are drawn in through the air suction. These sheets of water are estimated to have a thickness of from four- to eight-thousandths of an inch, and have a tendency to enlarge as the throat of the converging cone is approached. A high velocity and compact column of water and gases is produced in the throat of the converging cone. A diverging cone is fitted to the converging cone, and the kinetic energy at the throat section is transformed into work of compression in the diverging cone, thus overcoming the atmospheric pressure.

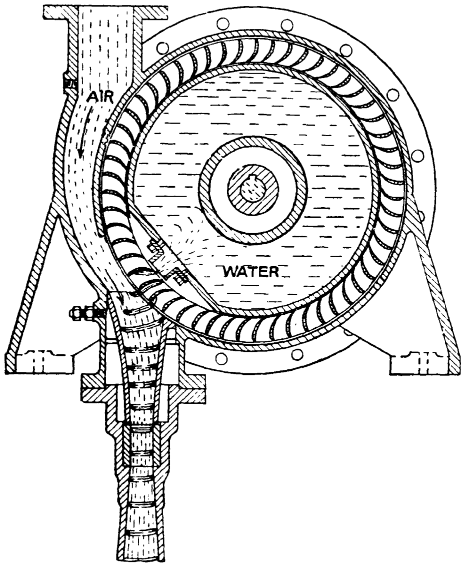


Fig. 75.—Mirrlees-Le Blanc Rotary Air Pump

As was stated on p. 93, one of the advantages of this pump is its inherent ability to condense out a quantity of the vapour which is associated with the air and non-condensable gases. The quantity of vapour condensed out depends on the temperature and quantity of sealing water necessary for the efficient operation of the pump; also on the quantity of vapour associated with the air, which latter depends on the difference between the vacuum temperature and the actual temperature of the air and vapour, or in other words, on the partial air pressure. This pump when supplied with cold sealing water possesses in itself an air cooler and devaporizer, which is a very valuable property as it adds greatly to the air dealing capacity of the apparatus.

Fig. 76 shows the air handling capacity characteristics of this type of air pump, and the power required to drive the pump at the varying air pressures.

It should be noted that the pressures on this curve are absolute air pressures, that is, the partial pressure of the air which is associated with the vapour. For example, take a 28 in. Hg vacuum, with sealing water at 80° F. with a barometer of 30 in. Hg. The vapour pressure at 80° F. is 28·93, so that the partial air pressure is 28·93 − 28, which equals 0·93 in. Hg. Descending from this air pressure to the capacity curve, the weight of air dealt with is read on the left-hand scale, i.e. 95 lb. per hour. This means that the pump will handle 95 lb. per hour of external air, which has leaked into the vacuum system, when supplied with sealing water at 80° F., and maintain a vacuum of 28 in. Hg. At colder sealing water temperatures the vacuum would be higher when the pump was dealing with the same air quantity, and the increase would correspond to the lower vapour pressure of the colder sealing water. If we descend from the air pressure of 0·93 in. Hg to the power curve, we read the b.h.p. absorbed by the curve to be 55.

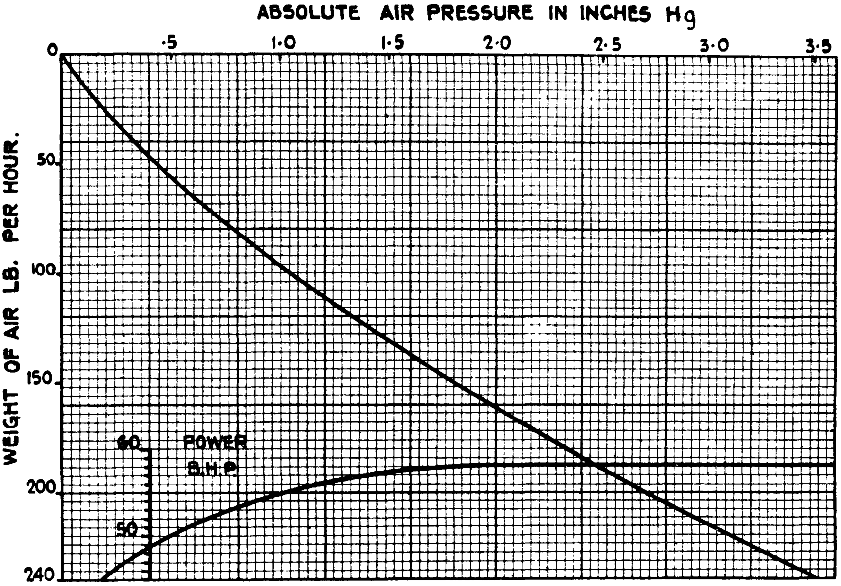


Fig. 76.—Mirrlees-Le Blanc Rotary Air Pump, Air Handling Capacity and Power Curves

The quantity of sealing water required by these pumps is approximately 1 gall. per minute for each b.h.p. required to drive the pump, and the peripheral speed of the blade wheel rim is about 88 ft. per second.

AIR PUMP CAPACITY TESTS AND AIR LEAK NOZZLES

The capacity of an air pump is measured by the weight or volume of free air which the pump is capable of handling when maintaining a specific vacuum.

The usual method of measuring the air capacity is by means of air leakage nozzles. The air is allowed to flow through these nozzles into the space being exhausted, and the quantity entering depends on the air temperature, the height of the barometer, and the throat diameter of the nozzles. The performance of the pump is usually set out as a curve plotted from a series of readings taken of vacuum attained against the weight of air handled, as calculated from the throat diameter of the air leak nozzles used. (See performance curves of air ejectors and air pumps, pp. 145 to 154.)

Two methods of testing air pumps may be employed; one is known as the dry air capacity method, and the other as the saturated air capacity method. The former consists of fitting a blank flange to the air pump suction and fitting the leak nozzle and a connection for a mercury column to this blank flange. The vacua recorded, corrected to the barometer, is noted for various air leak nozzles and a curve plotted, which is taken as the dry air performance. The latter method consists of connecting the air pump suction to a closed vessel or tank. A water connection is fitted to the top of the tank, and is so arranged that the water is broken up into a spray as it enters; the air leak nozzle is fitted so that the air is intimately mixed with the water, and thus becomes saturated with vapour at the temperature which obtains in the tank. This saturated air is drawn off by the air pump, and from a series of readings of vacua and air leak nozzles a curve of performance is plotted for the water temperature in the tank. With kinetic apparatus, such as steam-jet operated ejectors, the former method always shows a greater air handling capacity; this is due to the associated vapour which is dealt with along with the air. For the extraction of 1 lb. of air the mass of fluid being dealt with in the former method is 1 lb., whereas, in the latter method, the mass is 1 lb. plus the mass of the associated vapour. In both cases the volume per pound of air is the same at the same temperature and the same air pressure, but the total absolute pressures will be different by the amount of the vapour pressure at that temperature. With displacement apparatus, such as reciprocating air pumps, it therefore follows that, for equal weights of air dealt with, the vacuum obtained during a test carried out by the former method will be higher than that attained during a test carried out by the latter method by the amount of the vapour pressure. In other words, if, during a dry air test of a displacement air pump, water at a certain temperature be admitted to the vessel being exhausted, the vacuum will fall by an amount equal to the vapour pressure at that temperature.

With an air pump of the kinetic type it is the mass of fluid being handled that determines the vacuum, independent of the relative proportions of the air and vapour. This can be proved if an air pump of this type be first tested under dry air conditions and then tested under saturated air conditions, and curves of performance of each test plotted on the same diagram. It will be found from these curves that for any vacuum the weight of fluid handled in each case is very approximately the same.

The curves given in fig. 17 (p. 40) are useful in finding the mass of vapour which is associated with 1 lb. of air at various vacua and temperatures (see also pp. 37 to 41).

TO FIND THE QUANTITY OF AIR FLOWING THROUGH AN AIR NOZZLE

Let fig. 77 represent the nozzle with the flow as indicated.

Let P_1, V_1, and T_1 be the absolute pressure, volume, and absolute temperature of the air entering; and
P_2, V_2, and T_2, the absolute pressure, volume, and absolute temperature of the air at the throat section.

It is customary to use nozzles of the convergent type, thereby eliminating any uncertainty about the coefficient of contraction which with this type of nozzle may be taken as unity.

The expansion through the nozzle may be taken to be adiabatic with an exponent of 1·3. The exponent for pure dry air is 1·414, but atmospheric air has normally a percentage of vapour associated with it which reduces the exponent.

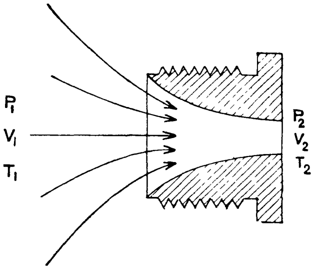


Fig. 77.—Air Nozzle

Assuming then an exponent of 1·3, we find the ratio of the throat pressure to the initial pressure to be

$$\begin{aligned} P_2/P_1 &= \{(1+n)/2\}^{n/(1-n)} \\ &= \{(1+1{\cdot}3)/2\}^{1{\cdot}3/(1-1{\cdot}3)} \\ &= (1{\cdot}15)^{-4{\cdot}33} \\ &= 0{\cdot}546. \end{aligned}$$

$$\text{Also} \quad P_1V_1^n = P_2V_2^n,$$

$$\text{or} \quad V_2 = (\sqrt[n]{P_1/P_2})V_1.$$

The velocity at the throat section is

$$v = \sqrt{2g\{n/(n-1)\}\{1-(P_2/P_1)^{(n-1)/n}\}P_1V_1}.$$

If a be the area of the throat section in square feet we get the volume flowing to be

$$V = v \times a \text{ c. ft. per second,}$$

and the mass flowing will be

$$m = v \times a/V_2 \text{ lb. per second,}$$

which by substitution becomes

$$m = [a\sqrt{2g\{1\cdot3/(1\cdot3-1)\}\{1-0\cdot546^{0\cdot2308}\}P_1V_1}]/(\sqrt[1\cdot3]{1\cdot832})V_1,$$

or $$m = (a\sqrt{36\cdot5P_1V_1})/(1\cdot593V_1).$$

P_1 and V_1 depend upon the reading of the barometer and the air temperature and the air humidity at the time of the test. The characteristic equation for pure dry air is PV/T = constant, and for pure dry air the value of the constant is 53·3. For normal values of the humidity no serious error is involved by neglecting it, so that we may write $P_1V_1/T_1 = 53\cdot3$, where T_1 is the absolute temperature in degrees Fahrenheit of the air entering the nozzle.

If H be the height of the barometer in inches of mercury then

$$P_1 = (H \times 14\cdot7 \times 144)/30,$$

and $$V_1 = (53\cdot3T_1)/(70\cdot6H).$$

Then, substituting these values of P_1 and V_1 in the equation for the mass passing per second, we get

$$\begin{aligned} m &= [a\sqrt{(36\cdot5 \times 70\cdot6H \times 53\cdot3T_1)/(70\cdot6H)}]/[(1\cdot593 \times 53\cdot3T_1)/(70\cdot6H)] \\ &= (36\cdot6aH)/\sqrt{T_1}. \end{aligned}$$

If d be the throat diameter of the nozzle in inches, the mass of air passing the nozzle in pounds per hour is

$$\begin{aligned} m &= \{(36\cdot6 \times 0\cdot7854 \times 3600)/144\}\{(d^2H)/\sqrt{T_1}\} \\ &= (719d^2H)/\sqrt{T_1}. \end{aligned}$$

If D be the throat diameter in millimetres, then

$$m = (1\cdot116D^2H)/\sqrt{T_1}.$$

The rule given by the B.E.A.M.A. for their standard air nozzle is

$$m = (1\cdot124D^2H)/\sqrt{T_1},$$

where D is the throat diameter in millimetres, and H the height of the barometer in inches of mercury, and T_1 the absolute temperature in degrees Fahrenheit of the air entering the nozzle, and the coefficient of contraction allowed is 0·98.

ALLOWABLE AIR LEAKAGE

On p. 37 the B.E.A.M.A. rule for the allowable air leakage on land surface condensing plants was given. This was

$$m = \{(0\cdot5W)/1000\} + 3 \text{ lb. dry air per hour.}$$

The above air quantity is for H.P. turbines, and for M.P. and L.P.

turbines it should be multiplied by 1·5, while for reciprocating engines it should be multiplied by 2·5.

Manufacturers now generally subject their air pumps to a capacity test, and for the purposes of such tests the B.E.A.M.A. have issued the following statement:

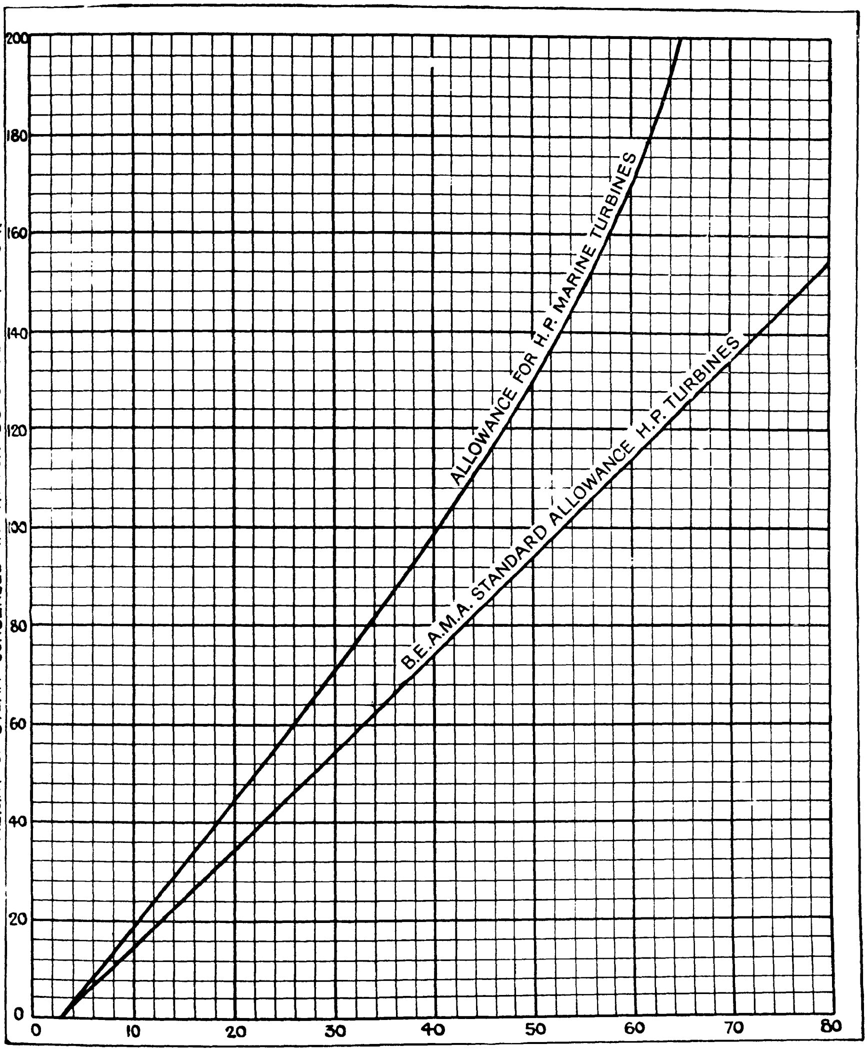


Fig. 78.—Curves of Allowable Air Leakage: Surface Condenser

The air pump connected to a closed vessel must be capable of maintaining in the closed vessel the specified vacuum when handling the above quantity of dry air admitted through a standard air nozzle, this air being saturated with water vapour by water at a temperature 5° F. above the inlet cooling water temperature, the water being sprayed into the closed vessel in quantities sufficient to saturate the air.

The rules issued by the B.E.A.M.A. apply particularly to land condensing plant, and, so far as the author is aware, no authorities have issued similar rules for marine condensing plants; the allowance for air leakage has been left entirely in the hands of the air pump manufacturers.

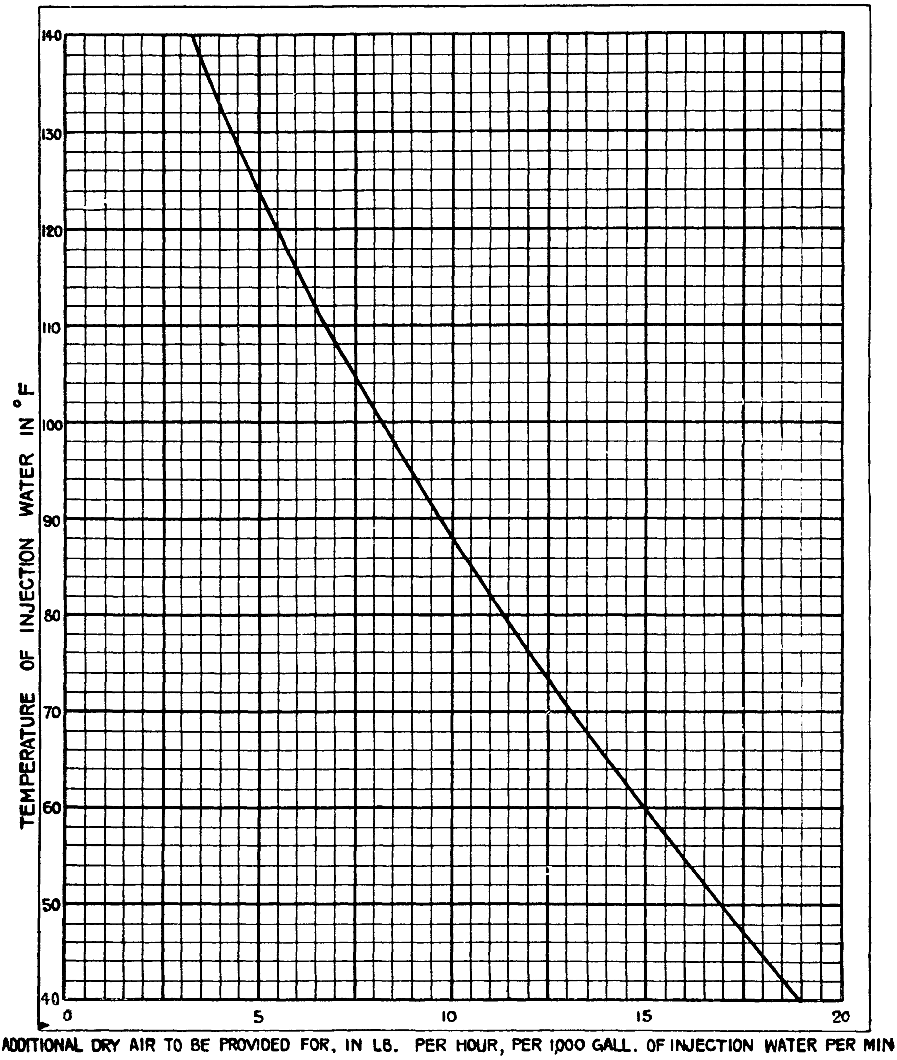


Fig. 79.—Curves of Allowable Air Leakage: Jet Condenser

During the author's experience with marine installations, and particularly with H.P. turbines of the Brown Curtis or Parsons designs, the average air leakage seldom exceeds two-thirds of that given by the above rule.

In jet condensers the air which enters with the injection water should be allowed for in addition to that which enters by leakage, and this addition may be taken at 15 lb. per hour per 1000 gall. of injection water per minute at 60° F.

The author has prepared the curves shown in figs. 78 and 79 which provide ample margin for air leakage. Fig. 78 gives the dry air to be provided for in surface condensers for varying weights of steam condensed. With jet condensers a percentage of air enters with the injection water, this air in addition to that which enters by leakage and that which comes over with the steam has to be handled by the air pump, so that a larger air pumping capacity must be provided for with jet condensing plants than with surface condensing plants. Fig. 79 gives the quantity of this additional air to be provided for varying the inlet temperatures of injection water.

The air enters as entrained air, and the quantity varies in accordance with the coefficient of absorption at each temperature, and with the mechanical treatment the water has received before entering the condenser; that is, water which is drawn from a pond is found to have less air in it than water drawn from a fast flowing river. The air to be provided for then in a jet condenser is the sum of that given in figs. 78 and 79.

For example, a land surface condensing plant, dealing with 50,000 lb. of exhaust steam per hour and maintaining a vacuum of 28 in. Hg with 4880 gall. per minute of cooling water at 80° F., would require an air pump having an air handling capacity of 28 lb. per hour (fig. 78), while a jet condensing plant working under the same conditions would require an air pump having a capacity of (fig. 79)

$$
\begin{aligned}
&= 28 + (11{\cdot}3 \times 4880)/1000 \text{ lb. per hour} \\
&= 28 + 55{\cdot}4 \\
&= 83{\cdot}14 \text{ lb. per hour.}
\end{aligned}
$$

These air quantities would be measured at a temperature of 85° F., the air being saturated with water vapour at that temperature.

INVESTIGATION OF THE REQUIREMENTS OF AIR PUMPS FOR HIGH AND LOW VACUUM

For the purpose of this investigation we shall take the high vacuum as 28·5 in. Hg, and the low vacuum as 26 in. Hg.

Let us take two condensers, one operating at 26 in. Hg vacuum and the other operating at 28·5 in. Hg vacuum, these vacua being recorded at the bottom of the condensers. In each case let there be a temperature depression of the condensate of 10° F., i.e. the condensate temperature is 10° F. lower than the vacuum temperature, due to the presence of air. In the low vacuum condenser then the condensate temperature will be 126° F. — 10° F. = 116° F., and in the high vacuum condenser the condensate temperature will be, say, 92° F. — 10° F. = 82° F. The partial air pressure in the low vacuum condenser will be 26·93 — 26 = 0·93 in. Hg, and in the high vacuum condenser the partial air pressure will be 28·9 — 28·5 = 0·4 in. Hg. These figures are shown for comparison in fig. 80.

This means that, if an air pump be coupled to the low vacuum condenser, and at a certain speed it reduced the partial air pressure to 0·93 in. Hg, then if this same air pump be coupled to the high vacuum condenser the speed would have to be greatly increased to reduce the partial air pressure to 0·4 in. Hg, and deal with the same air leakage. In other words, this means that for the same temperature depression the air pump coupled to a low vacuum condenser need not be so large as the air pump coupled to a high vacuum condenser.

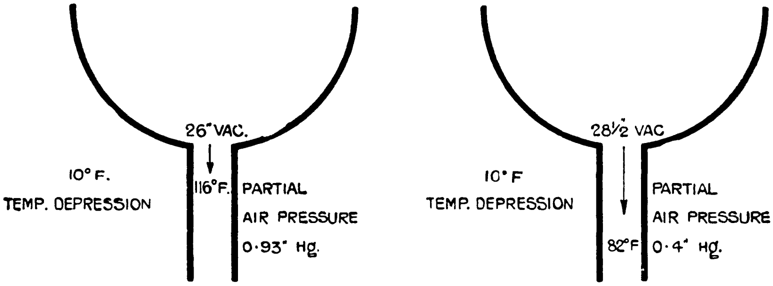


Fig. 80

Now let the partial air pressure in the high vacuum condenser be increased till it is the same as in the low vacuum condenser, i.e. 0·93 in. Hg; the temperature depression in the high vacuum condenser becomes 92 — 63 = 29° F., and the conditions will be as shown in fig. 81.

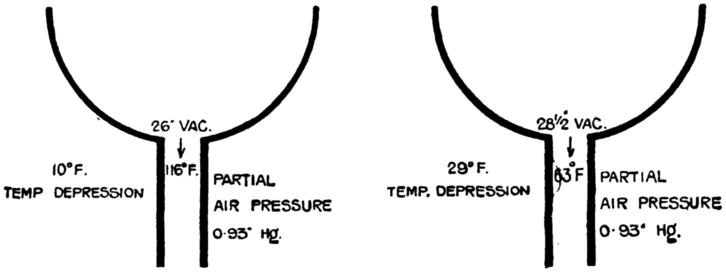


Fig. 81

The same air pump on these two condensers will, when running at speeds proportional to the absolute temperatures of the condensate and when dealing with the same air leakage, give a temperature depression in the low vacuum condenser of 10° F., and in the high vacuum condensers, 29° F. The change in the speeds of the air pump is due to the reduced volume of the air in the high vacuum condenser, and the ratio in this case of a displacement pump is

$$(459{\cdot}6 + 116)/(459{\cdot}6 + 63)$$
$$= 1{\cdot}1 \text{ to } 1.$$

The temperature depression is a direct loss of heat, the heat passing away in the condenser circulating water. This loss must be made up by increased coal consumption in the boilers, and should be reduced to a minimum. A temperature depression of 29° F., as shown above, represents about 2·9 per cent of the coal consumption. A simple displacement air pump therefore which gives a loss of 1 per cent at 26 in. Hg vacuum would give a loss of 2·9 per cent at $28\frac{1}{2}$ in. Hg vacuum, and a larger loss at higher vacuum. The conclusion, therefore, is that, although the simple displacement air pump is quite well suited for and is the proper apparatus for low vacuum, the loss becomes excessive when it is applied to high vacuum. This fact led engineers to look around for alternative means of creating high vacuum, which resulted in the steam-jet augmentor being applied to displacement air pumps, steam-jet operated air ejectors, water-jet operated air ejectors, and in the Weir dual air pump.

MARINE AIR PUMPS

The present-day air pump used on board ship is generally of the displacement type. With the advent of the geared turbine, however, a call came for higher vacuum, because with this type of prime mover it was possible, owing to its reduced dimensions, to provide sufficient space for the larger low pressure blading necessary to take advantage of a higher vacuum. Even with geared turbines, however, a vacuum higher than 28·5 in. Hg is seldom called for, due primarily to considerations of the weight and space occupied by the larger low pressure turbine and condenser. The maximum vacuum for marine conditions may be taken therefore to be 28·5 in. Hg. To meet this condition requires a special air pump which, though still of the displacement type, has provisions made for the cooling and devaporizing of the air quite independent of the condensate temperature. This air cooling and devaporizing greatly reduces the air volume, and brings the air pump within reasonable dimensions. This may be clearly shown by the following example.

Let us find the displacement (*Case 1*) necessary with an ordinary air pump of either the Edwards or the three-valve type to maintain a vacuum of 28·5 in. Hg with the barometer at 30 in. Hg, against an air leakage of 30 lb. per hour, the temperature of the condensate being 80° F.

Let us thereafter (*Case 2*) find the necessary displacement when the air is cooled to 70° F. and compare the displacements.

Case 1.—The partial pressure of the air entering the pump suction is

$$\begin{aligned} &28{\cdot}97 - 28{\cdot}5 \\ &= 0{\cdot}47 \text{ in. Hg.} \end{aligned}$$

The volume occupied by 1 lb. of air at this pressure and 80° F. is

$$\begin{aligned} V_1 &= \{12{\cdot}39 \times 30(459{\cdot}6 + 80)\}/\{0{\cdot}47(459{\cdot}6 + 32)\} \\ &= 868 \text{ c. ft. per pound of air.} \end{aligned}$$

The total volume to be dealt with per hour by the air pump is

$$= 30 \times 868$$
$$= 26{,}040 \text{ c. ft.}$$

Case 2.—The partial pressure of the air entering the pump suction is

$$= 29{\cdot}26 - 28{\cdot}5$$
$$= 0{\cdot}76 \text{ in. Hg.}$$

The volume occupied by 1 lb. of air at this pressure and 70° F. is

$$V_2 = \{12{\cdot}39 \times 30(459{\cdot}6 + 70)\}/\{0{\cdot}76(459{\cdot}6 + 32)\}$$
$$= 520 \text{ c. ft. per pound.}$$

The total volume to be dealt with by the air pump per hour is

$$= 30 \times 520$$
$$= 15{,}600 \text{ c. ft.}$$

The ratio of the displacements is

$$26040/15600$$
$$= 1{\cdot}669,$$

i.e. the air pump without the air cooling must be 1·669 times larger than with the air cooling.

The function of an air pump is to remove air and condensate from a steam condenser with the minimum expenditure of energy, the air being removed at as low a temperature as possible, and the condensate being removed at as high a temperature as possible, i.e. the vacuum temperature. This desideratum indicates that the air and water must be dealt with separately, the air being cooled either before it enters the air pump or inside the pump barrel, while the water enters the pump direct.

A consideration of the above facts shows why the modern air pump of the reciprocating type and installed for high vacuum utilizes two barrels, one dealing with the condensate only, the other with the air and vapour only. These barrels are generally termed the wet and dry barrels respectively.

The air pump installed on turbine-driven vessels is invariably of the independent type, i.e. it is not driven from the main turbine but is driven by its own steam cylinder, or, in some very modern ships, by its own electric motor.

The reasons for the independent pump may be briefly stated as follows:

The vacuum can be created in the main condenser before the turbine is started, thus preventing excessive heating and cooling of L.P. turbine and condenser, and so relieving them of heavy temperature stresses every time the turbine is started or stopped.

Being independent of the main turbine the air pump can be run fast or slow, depending upon the load on the main turbine and upon the air leakage.

For rapid and easy manœuvring the independent air pump is indispensable, since a good vacuum must be maintained in the condenser for a turbine to give a rapid and certain response to the manœuvring valve, this is of especial importance since the safety of the ship may depend on the speed of manœuvring.

The independent air pump may be taken to be indispensable to all classes of turbine-driven ships and to reciprocating-engined ships with main engines of 5000 i.h.p. and over.

The Weir independent dual air pump, which is largely used on board ship in conjunction with turbine machinery, is shown in figs. 82 and 83. This air pump embodies much of the foregoing desiderata in a convenient and compact and efficient manner.

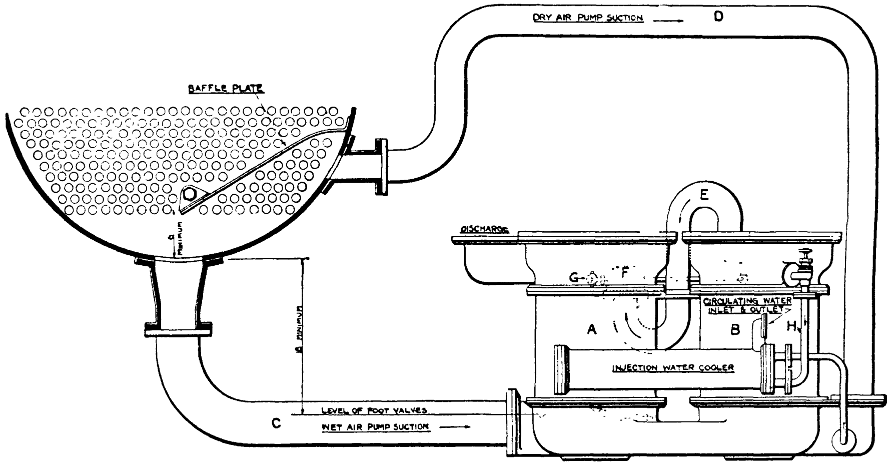


Fig. 82.—Weir Condenser System

Diagrammatic Arrangement of Connections between Dual Air Pump and Condenser

A is the wet barrel and B is the dry barrel.

The wet barrel A is connected to the bottom of the condenser by the suction pipe C, all the condensate falls to the bottom of the condenser and flows by this pipe to the wet pump A. D is the dry air pump suction through which the dry barrel B draws the air, vapour, and non-condensable gases. Both barrels are normally of the three-valve type, i.e. with foot, bucket, and head valves. One novel feature of this pump is the method adopted for dealing with the discharge from the dry pump B. The pump B discharges through the discharge pipe E and thence through a spring-loaded valve F, the air and vapour entering the wet pump through a port which is situated just under the head valves of the wet pump. By this arrangement the load on the dry pump is greatly reduced, because the pressure against which the dry pump discharges is reduced from atmospheric pressure, if the pump had discharged to atmosphere, to 20 in. Hg vacuum. Both methods of cooling the air are shown in figs. 82 and 83, i.e. air cooling external to the pump and air cooling inside the pump barrel. The former method consists of fitting an air cooler integral with the main condenser as shown, and as previously described; the latter method is by the use of a small injection cooler placed alongside the pump. This cooler is of tubular construction and employs two water circulation circuits, one inside the tubes and one outside the tubes. The one is a cold sea-water circulation taken from the main circulating pump, the other is a circulation of fresh water and is maintained by the working of the dry pump. For the freshwater circulation a pipe H is led from the dry pump discharge to the bottom cooler connection; from the top cooler connection a pipe is led to the pump suction branch. The relatively hot water discharged by the dry pump flows down the pipe H, and passing through the cooler and inside the tubes becomes cooled by the cold sea-water circulating outside the tubes. This cooled water entering the dry suction branch meets the entering air and vapour and reduces their volume by condensing out a portion of the vapour and by lowering the temperature. This results in the temperature of the dry barrel running from 10° F. to 12° F. lower than the temperature of the wet barrel. To make up for loss of sealing water a small pipe G with valve is fitted between the wet and dry pump, so that any water which may be discharged over the discharge pipe E may be made up by the opening of the small make-up valve. This type of pump gives the air a two-stage compression: the first compression is given in the dry pump and is from the partial air pressure at the dry suction branch to 20 in. Hg vacuum the dry pump discharge pressure; the second compression is given in the wet pump and between the bucket and the head valves, the air being compressed and discharged to atmosphere on the up stroke of the wet pump.

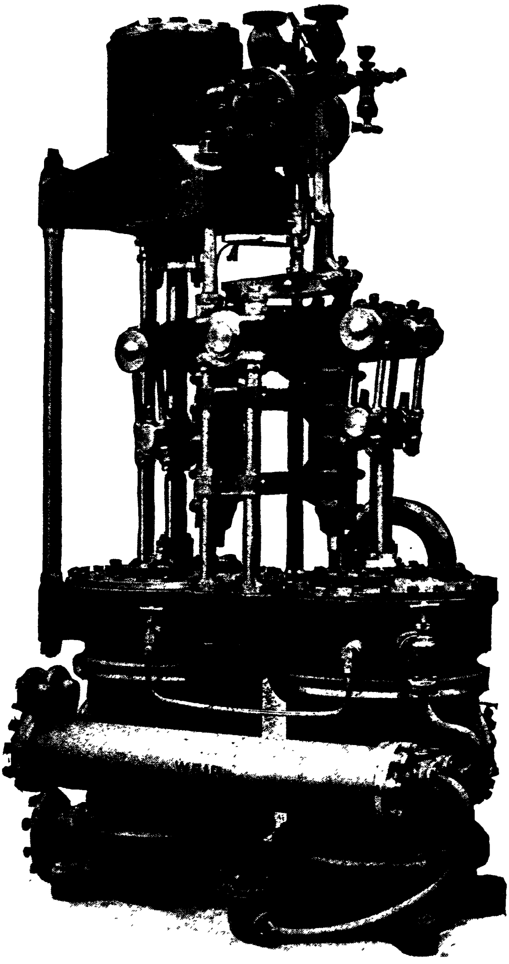


Fig. 83.—Weir Independent Dual Air Pump

The Weir dual air pump when installed on board ship is generally steam driven. The steam cylinder is arranged above the wet pump, and the dry pump is driven by levers, the arrangement being as shown in fig. 83.

This pump is very economical in steam, the consumption being approximately 1 per cent of the total quantity of condensate being handled. The economy is primarily due to the disposition of the steam cylinder over the wet pump, coupled with the fact of the dry pump discharge pressure being 20 in. Hg vacuum, and the arrangement of the throttle ports on the valve faces of the steam chest. On the up stroke of the wet pump the steam port opening is large, and the wet pump is thus able to discharge its contents against the pressure of the atmosphere. On the down stroke of the wet pump there is little resistance from the pump, because the contents are simply transferred from one side of the bucket to the other through the bucket valves. Meantime, the dry pump is on the up stroke, and having only to discharge its contents against a vacuum of 20 in. Hg offers little resistance. On the down stroke of the piston, therefore, there is little resistance to be overcome, and on this stroke the steam supply to the cylinder is restricted by being throttled through a small port on the steam valve face. This results in the steam pressure acting on the top of the piston being greatly reduced, so that the quantity of steam in the cylinder at the end of the down stroke is much less than the quantity at the end of the up stroke, the difference being a direct saving. The arrangement further provides a reasonably uniform speed of up and down strokes.

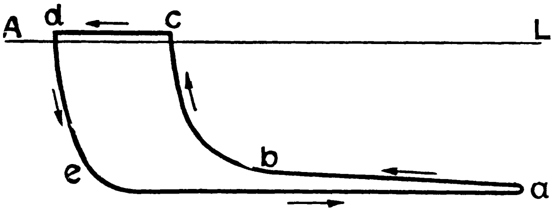


Fig. 84.—Indicator Card taken from the Wet Pump of a Weir Dual Air Pump

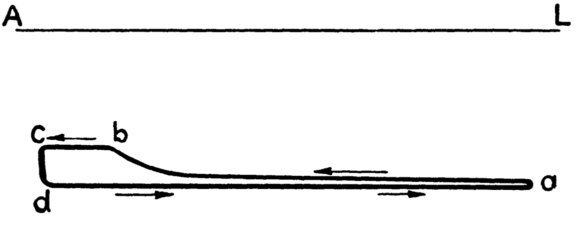


Fig. 85.—Indicator Card taken from the Dry Pump of a Weir Dual Air Pump

Fig. 84 shows an indicator card taken from the wet pump of a Weir dual air pump and from above the bucket.

Fig. 85 shows an indicator card taken from the dry pump of a Weir dual air pump and from above the bucket.

Fig. 86 shows an indicator card taken from the steam cylinder.

In fig. 84 taking the bucket at the bottom of its stroke the compression starts at *a*, and the compression line *abc* is very nearly isothermal, due to the compression taking place in the presence of water; from *c* to *d* discharge takes place under constant pressure. From *d* to *e* there is a rapid fall in pressure, owing to the water in the clearance space above the bucket leaving the head valves. When the pressure has fallen sufficiently to overcome the spring load of the dual valve, this valve opens and the contents of the dry pump are discharged into the wet pump along the line *ea*.

In fig. 85 the air is compressed isothermally from *a*, and when the vacuum is reduced to a little over 20 in. Hg, the head valves open and discharge takes place from *b* to *c*, the return stroke then takes place and the pressure falls to *d*, the return stroke being completed from *d* to *a*.

In fig. 86 the full line gives the card from above the piston, while the dotted line gives the card from the underside of the piston. In the full line card when the steam valve opens the pressure on the underside of the piston rises from *h* to *a*, and the piston starts ascending. From *a* to *b* the pressure remains sensibly constant, while from *b* to *c* the pressure rises to the steam supply pressure, owing to the piston being slowed down by the increasing load from the wet pump discharging; from *c* to *d* the pressure remains constant and the wet pump discharge is completed. The steam valve is shot at *d*, and the bottom of the cylinder is put into communication with the exhaust and the pressure falls to *e*; the piston then starts its downward stroke and travels relatively fast from *e* to *f*, where it again slows up owing to the dry pump discharging and the pressure falls to *g* and continues at the exhaust pressure to *h*, where the steam valve is again shot and the cycle is repeated.

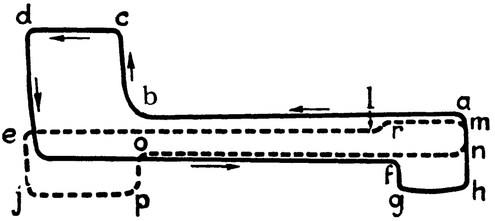


Fig. 86.—Indicator Card from the Steam Cylinder Weir Dual Air Pump

In the dotted card the steam pressure rises from *j* to *e*, and the piston starts to travel downwards to *l*, where, owing to the slowing up, the pressure rises to *r* and continues to *m*. At *m* the steam valve is shot and the top of the cylinder is put into communication with the exhaust and the pressure falls to *n*, the piston then starts ascending and ascends relatively fast to *o*, where it slows up and the pressure falls to *p*, the exhaust pressure, and continues at exhaust pressure to *j*, where the steam valve again shoots and the cycle is repeated.

Table V, p. 171, gives particulars from the test sheet of a Weir dual air pump, which were taken in conjunction with the foregoing indicator cards; this table indicates the performance of these pumps.

From these figures the effect of the cooler is clearly shown. This pump will handle 33·3% more air and maintain the same vacuum than will a normal displacement air pump of the same dimensions and running at the same speed.

From equation (32) the vapour condensed out works out to be 13·44 lb. per hour; the temperature rise in the cooling water flowing through the cooler is 4° F., therefore to absorb the heat given out by the above quantity of vapour condensing at 1000 B.Th.U. per pound, the quantity of cooling water flowing through the cooler must have been 5·6 gall. per minute.

TABLE V

Load in pounds of water per hour	280,000	137,500
Barometer, in. Hg	30	30
Vacuum recorded, in. Hg	28·65	27·8
Vacuum correct to 30 in. Hg barometer	28·65	27·8
Vacuum at dry pump discharge, in. Hg	20	20
Sea temperature, degrees F.	61	61
Wet pump discharge temperature, degrees F.	90	90
Dry pump discharge temperature, degrees F.	73	72
Sealing water temperature, degrees F. Inlet to cooler	73	72
Sealing water temperature, degrees F. Outlet from cooler	69	68
Circulating water temperature, degrees F. Inlet to cooler	61	61
Circulating water temperature, degrees F. Outlet to cooler	63	63
Speed of pump (strokes per minute)	54	54
Steam supply pressure, pounds per square inch	125	110
Air leakage in pounds per hour	0	34

DISPLACEMENT AIR PUMPS WITH STEAM JET AUGMENTERS

The first really practical vacuum augmenter to be put on the market was that introduced by the Hon. C. A. Parsons. This device is shown in fig. 87.

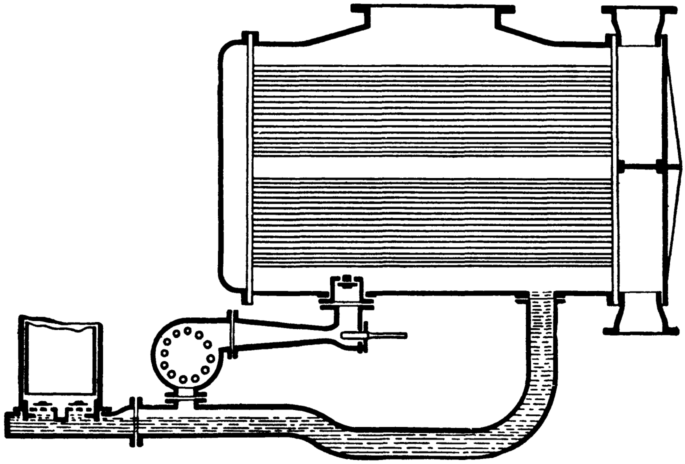


Fig. 87.—Parsons' Vacuum Augmenter

Two suctions are fitted to the main condenser, one dealing with the condensate and the other dealing with the air and vapour. The augmenter is fitted to the latter suction, and is simply the first stage of a two-stage air ejector. The augmenter discharges to a small augmenter condenser where the steam used by the augmenter is condensed. The vacuum carried in the augmenter condenser varies with the air being handled, but normally runs from 26 to 27 in. with 28·5 in. Hg vacuum in the main condenser. The air and vapour receive their first compression in the augmenter, so that the air pump is only called upon to deal with these at the low vacuum of 26 to 27 in. Hg. This greatly increases the air-handling capacity of the air pump, and consequently, when an augmenter is fitted to an air pump, the size of the pump installed can be made much smaller than if no augmenter be fitted.

The wet suction pipe is looped, as shown, to provide a water seal and ensure that the air and non-condensable gases which have already been compressed by the augmenter do not flow black along the wet suction pipe to the condenser, thus creating an internal circulation which destroys the efficiency and operation of the apparatus. Owing to the higher vacuum carried in the main condenser, the condensate will stand in the wet suction pipe at a higher level than that in the air pump suction chamber. The difference in levels will be directly proportional to the difference of vacuum in the main condenser and in the air pump suction chamber. If the vacuum in the main condenser is 28·5 in. Hg, and the vacuum in the pump suction chamber is 26 in. Hg, then the difference in the water levels will be

$$\{(28{\cdot}5 - 26)34\}/30 \text{ ft. of water} = 2{\cdot}92 \text{ ft.}$$

It is very important that sufficient head be provided from the bottom of the main condenser to the water level in the pump suction chamber to ensure that the high level is not up inside the main condenser and the augmenter suction branch drowned. Sufficient head is sometimes very difficult to arrange on board ship, especially with the modern underhung condensers, and the writer knows of cases where the vacuum augmenter gives no increase in vacuum due solely to the above cause. The steam used by the jet is additional to that used by the air pump. Turbine machinery is, however, very sensitive to vacuum, and the gain of $\frac{1}{2}$ in. so reduces the steam consumption of the main turbines, that there is a net saving. It should be clearly borne in mind, however, that for a turbine to benefit by a higher vacuum it should be designed for the higher vacuum; no great reduction in steam consumption need be anticipated by running a turbine with a vacuum higher than that for which it was designed. This was clearly demonstrated some years ago, when at the instigation of Mr. William (now Lord) Weir, the British Admiralty conducted some trials on torpedo-boat destroyers at reduced vacua. The results of these trials showed that in every case when the vacuum was reduced to the designed vacuum the overall economy rose.

The following gives Baumann's published figures for the decrease in steam consumption for various increases in vacua:

4 per cent decreased consumption per 1 in. Hg vacuum between 26 and 27 in. Hg.

5 per cent decreased consumption per 1 in. Hg vacuum between 27 and 28 in. Hg.

6 per cent decreased consumption per 1 in. Hg vacuum between 28 and 29 in. Hg.

Messrs. G. & J. Weir, Ltd., have put on the market for marine purposes a new design of air pump with steam jet augmenter, and called by them *The Paragon Air Pump*. This pump is shown in fig. 88, and is of the twin-barrel type, driven by a single steam cylinder.

Both barrels deal with the water, i.e. the suctions are common and each barrel draws its quota of condensate through the same suction pipe. The pumps are of the three-valve type with a normal design, with foot, suction, and head valves. On the side of each barrel, and high up on the barrel, is arranged a port A, so positioned that when the bucket is at the top of its stroke the port is in communication with the space under the bucket, and when the bucket has travelled on its down stroke, a distance equal to its own depth plus the depth of the port, the port is in communication with the space above the bucket. Communicating with this port A is a chamber B which houses a non-return plate valve C of a design similar to the pump valves. A branch pipe D connects the two chambers B with the augmenter condenser outlet, the augmenter condenser E is of the tubular type and fitted with condenser tubes ferruled into brass tubeplates to provide for expansion. The augmenter F is fitted to the top of the augmenter condenser and discharges directly into the same. The steam discharged by the augmenter is condensed on the outside of the tubes by the condensate discharged by the air pump. The heat in the augmenter discharge is therefore conserved in the condensate, which results in the economy of the apparatus being good. The steam

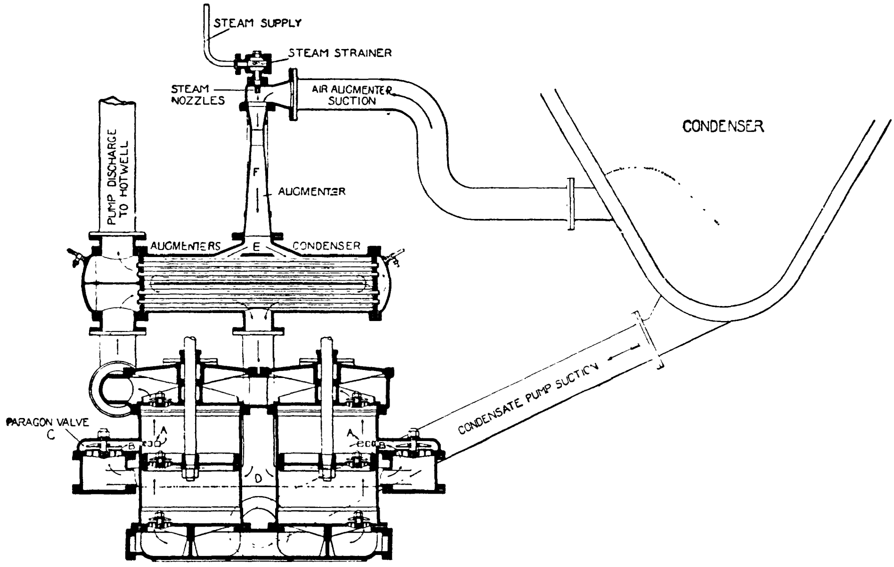


Fig. 88.—Weir Paragon Air Pump

nozzles are of the convergent-divergent type, and arranged as a group of small nozzles having a throat diameter of 3/32 in. A steam strainer is provided to prevent choking of the steam nozzles. The whole apparatus is compact and self-contained, and is thus easily installed on board a ship. Fig. 89 shows the characteristic performance of this pump, having barrels 19 in. diameter × 15 in. stroke, and the augmenter using 280 lb. of steam per hour. The curve A gives the vacuum curve for the pump alone, and the curve B gives the vacuum curve when the augmenter is turned on.

It should be noted that within the range of air leakage shown the increase in vacuum given by the augmenter increases with the air leakage. The curves A and B are plotted from figures taken with a condensate temperature of 80° F. Two other curves C and D are plotted from figures taken with a condensate temperature of 50° F. The maximum vacuum obtained on curve C at zero leakage is 29 in. Hg, while the turning on of the augmenter raised the vacuum to 29·63 in. Hg. The vacuum of 29 in. Hg obtained with the air pumps only working, gives the stalling point of the pump, i.e. the point at which a decrease in condensate temperature gives no increase in vacuum, and the difference between this vacuum and the vacuum of 29·63 in. Hg obtained when the augmenter was turned on represents the loss of vacuum due to the friction through the pump foot valves and grids, the weight of

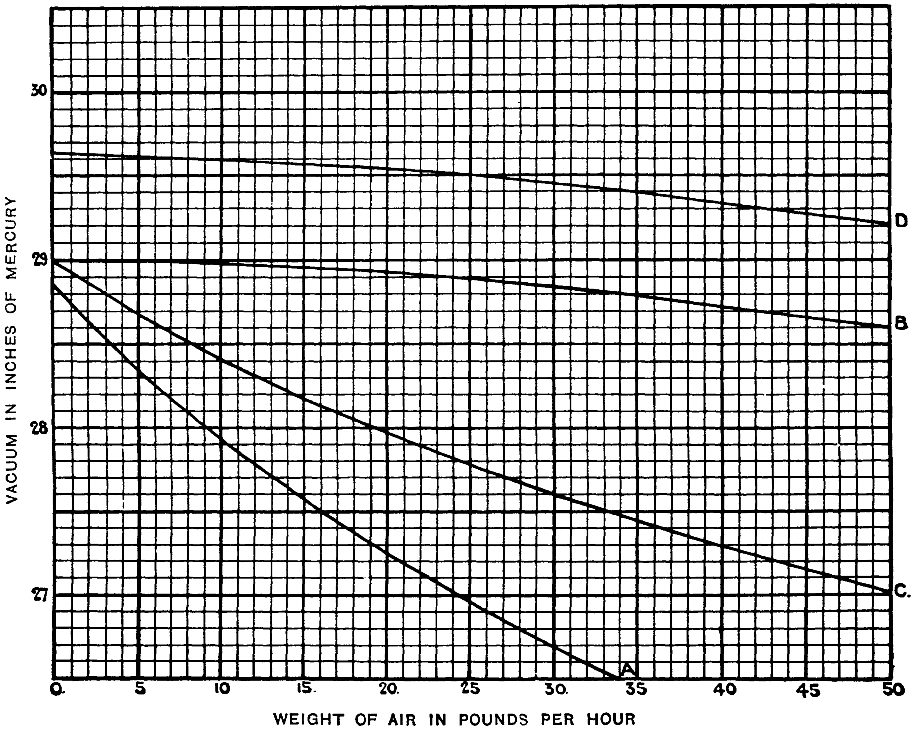


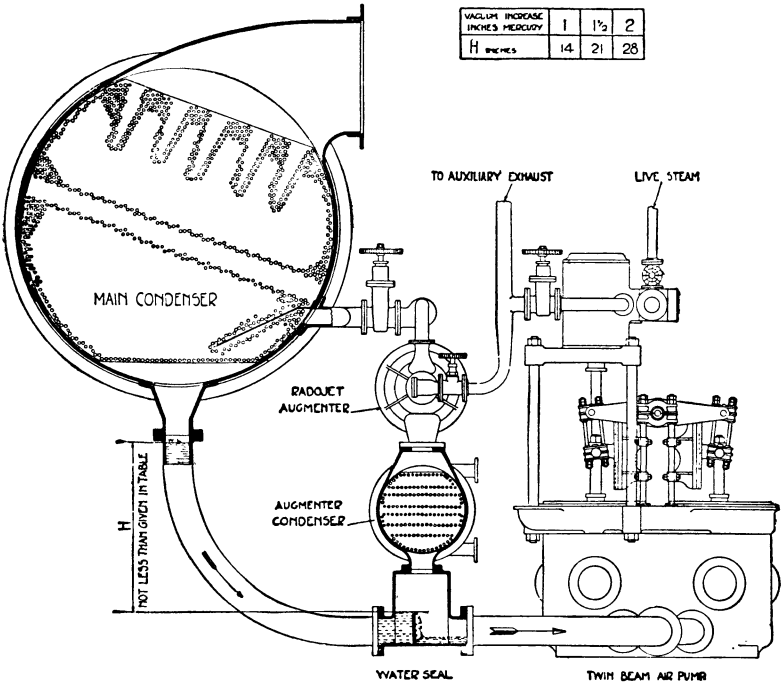
Fig. 89.—Vacuum Curves of Paragon Air Pump

the valves and the negative head caused by the water having to be drawn up into the pump barrel on the up stroke of the pump.

RADOJET VACUUM AUGMENTER

The arrangement adopted by Messrs. J. Musgrave & Co., Bolton, for their radojet vacuum augmenter is shown in fig. 90.

The augmenter is generally on the lines of their straight radojets, as illustrated and described on pp. 146 to 151. Only the second stage, however, is employed as an augmenter, and it is designed for operating with low pressure exhaust steam at 2 to 5 lb. per square inch gauge. As shown in fig. 90 the augmenter draws the air and non-condensable gases by a separate suction from the main condenser, and discharges them at an increased pressure to the tubular augmenter condenser, where the steam used by the augmenter is condensed, and the air and non-condensable gases pass to the water seal chamber arranged underneath the augmenter condenser. The function of the water seal is precisely the same as that described for Parsons augmenter. The air and non-condensable gases then pass at the reduced vacuum to the twin beam air pump, and are dealt with in the usual fashion. Live steam is supplied to the pump only, a branch being taken off the pump exhaust to supply the augmenter.



VACUUM INCREASE INCHES MERCURY	1	1½	2
H INCHES	14	21	28

Fig. 90.—Musgrave Radojet Condenser

The following table gives the performance of a Radojet augmenter fitted to the twin beam air pump on a foreign torpedo-boat destroyer running under full power.

	Vacuum in Condenser.	Steam Supply Pressure.	Sea Temperature.
Radojet augmenter in use	27·9 in. Hg	4·3 lb./square inch gauge	69° F.
Air pump alone	26·12 in. Hg	—	69° F.

The steam used by the augmenter was 0·26 per cent of the steam consumption of the main turbine, and the increased vacuum obtained resulted in a decrease in the steam consumption of the main turbine of 9·22 per cent.

KINETIC AIR PUMP (ROTARY SYSTEM)

The method employed for the removal of air with this type of apparatus consists of a steam jet air ejector working in series with a water-jet operated ejector. The steam jet air ejector forms the first stage and the water-jet operated ejector the second stage. The supply water for the water-jet operated ejector is supplied by an ordinary low-lift centrifugal pump. The water-jet operated ejector discharges to a tank from which the centrifugal pump draws, the same water being used over and over again.

Fig. 91 shows one arrangement of the system as applied to a land surface condensing plant. The steam jet ejector A draws the air and vapour via the suction pipe B and discharges to the auxiliary surface condenser C. The water extraction or condensate pump D withdraws the condensate from the condenser E and delivers through the auxiliary condenser C, thus condensing the steam discharged by the steam jet ejector A, and absorbing in the condensate the heat in the ejector discharge. The air and uncondensed vapour flow through the non-return valve F, and enter the suction chamber of the water-jet operated ejector G. The operating water for this ejector is delivered by the " kinetic " pump H through a strainer J to the water nozzles K and L. The water nozzle K is a central solid nozzle, while the water nozzle L is an annular one. The combined discharge from the nozzles K and L, together with the entrained air and vapour, is delivered through a divergent diffuser tube M, where the kinetic energy of the rapidly moving jets is converted into pressure energy, so that the combined mass can discharge against atmospheric pressure. The diffuser tube M is situated centrally in the tank N, and from this tank the kinetic pump H draws. Owing to the energy losses involved, and to the heat in the outlet from the auxiliary condenser C, the water circulating round the kinetic pump and water-jet operated ejector gradually heats up. This would be reflected in impaired vacuum, and to provide against this a valve O, float ball P, float-operated valve R, and circulating pipe S are provided. When the valve O is opened the condensate passes to the suction of the kinetic pump H, and is discharged to the water-jet operated ejector G, which in turn discharges it to the tank N. The water level therefore rises and raises the float ball P, thus opening the valve R, and water is drawn up the pipe S and into the top of the main condenser; this water falls down among the cold tubes and is therefore cooled. A circulation round the tank and main condenser, therefore, is set up, and is regulated by the valve O and the temperature of the water entering the ejector is thereby controlled.

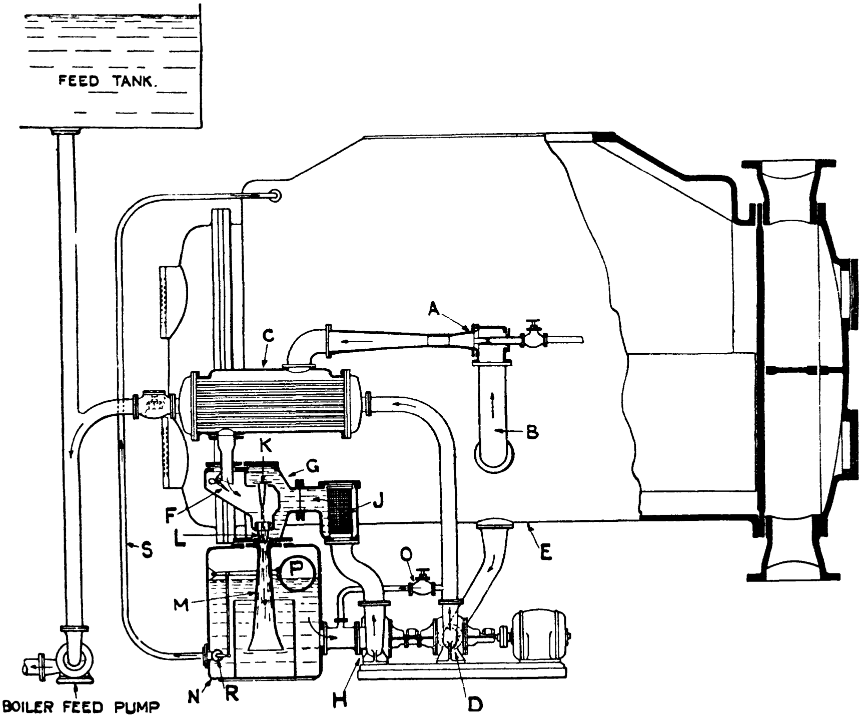


Fig. 91.—Kinetic Air Pump Arrangement

MARINE TYPE KINETIC AIR PUMP

As applied to ship work, the kinetic air pump is a combination of a reciprocating displacement pump with a steam jet augmenter. The air compression is accomplished in two stages, the steam jet taking the first stage and the displacement pump the second stage. Fig. 92 shows the arrangement in diagrammatic form.

The pump is of the twin-barrel type, the barrel A dealing with condensate only and drawing direct from the condenser, and the barrel B dealing with air and vapour. The augmenter J is connected by the branch C to the air-suction branch on the condenser, and discharges to the receiver D which is simply a small jet condenser. The steam discharged by the augmenter is condensed by condensate taken from the discharge of the wet pump A by the pipe E. The condensate is sprayed into the receiver D and condenses the steam by direct contact. The receiver is drained to the dry pump B by the pipe F, and enters the dry pump barrel through a non-return valve and port under the head valves. The augmenter J gives a compression to the air of 4 to 5 in. of mercury, depending on the air being handled, so that with 28·5 in. Hg vacuum in the main condenser the vacuum carried in the receiver would be about 24·5 in. Hg. This greatly reduces the volume of the gases to be dealt with by the dry barrel, and its capacity is therefore greatly increased. The dry barrel B draws the air and vapour from the receiver D by the pipe G.

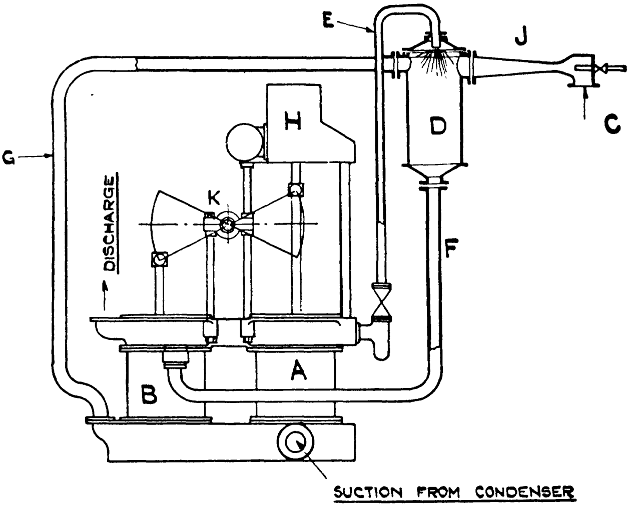


Fig. 92.—Arrangement of Marine Type of Kinetic Air Pump

Since the steam discharged by the augmenter is condensed by condensate which returns to the pump discharge, the heat used by the augmenter is absorbed by the feed water. The pump is driven by the steam cylinder H, the wet pump being directly driven by the piston rod and the dry pump being driven by the levers K from the wet pump crosshead.

EDWARDS AIR PUMP

Although the Edwards air pump is practically out of date for land power stations, no work such as the present would be complete without some sort of reference to it.

The main feature of the Edwards air pump is the mechanical injection of the condensate into the barrel and above the bucket, when the latter is at the bottom of its stroke, by the conical-shaped underside of the bucket. The action is illustrated in fig. 93.

When the bucket is performing about three-quarters of its upward and downward strokes, reckoned from top, the condensate flowing in by the suction branch B collects in the conical bottom of the pump chamber. The bucket descends on this water, and, because of the conical bottom given to

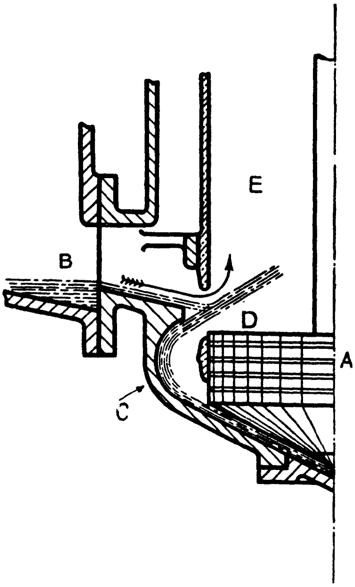


Fig. 93.—Edwards Air Pump

the bucket, it enters the water without shock and projects the latter round the curved portion C, which diverts the water through the ports D and into the barrel above the bucket. The leaving edge of the curved portion C is set at such an angle that the projected water passes through the port, so that a clear opening is left for the admission of air, as shown by the arrow, the rapidly moving projected water assisting the air to flow into the barrel. Normal head valves are fitted, and the descending bucket creates a vacuum in the barrel higher than that in the pump suction pipe. When the bucket has descended and uncovered the ports D, the barrel E is put into communication with the pump suction B and air is drawn into the barrel, the farther descent of the bucket causing the water to be projected into the barrel, and assisting in the entrance of the air. Owing to the top of the bucket being flat the working clearance between the bucket and head valves can be reduced to a minimum, so that

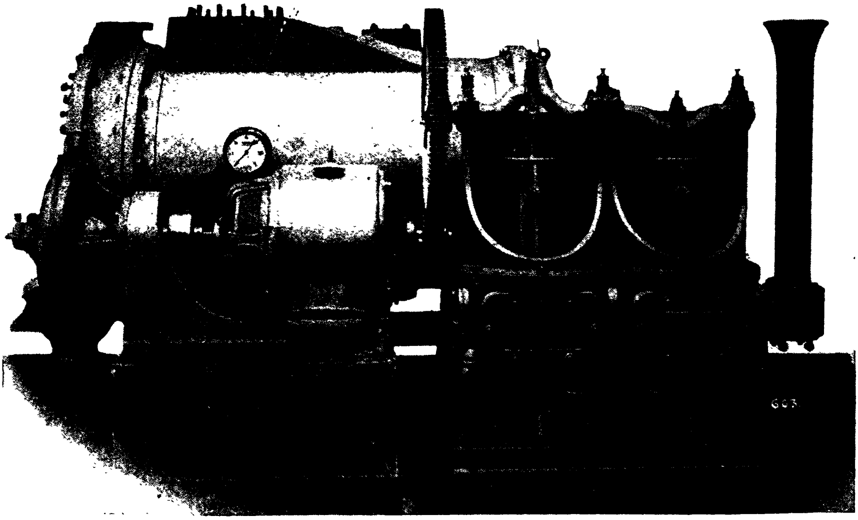


Fig. 94.—Two-throw Electrically Driven Edwards Air Pumps

the minimum quantity of air remains in the clearance space. This air, which is really in the nature of absorbed air, is released on the downward stroke of the bucket, owing to the reduction in absolute pressure. The air expands and exerts its own pressure at the end of the stroke, and consequently the vacuum in the barrel at the point of port opening is less than it would be were the clearance volume zero. The aim, therefore, is to reduce the clearance to a minimum, which can be readily done with this type of pump.

The Edwards air pump as installed on land is generally arranged with two or three barrels, all driven from a common crank-shaft. Fig. 94 shows a two-throw, pump motor driven through gearing as manufactured by Messrs. The Mirrlees Watson Company.

The air pump rating as recommended by the Edwards Air Pump Syndicate is as follows:

Vacuum, in. Hg.	Air Pump Displacement per Pound of Steam Condensed.
25	0·5 c. ft.
26	0·6 „
27	0·75 „
28	0·9 „

The Edwards air pump can be driven at speeds up to about 200 r.p.m., 150 r.p.m. being a common speed.

CHAPTER VII

Water-Extraction Pumps

MIRRLEES EXTRACTION PUMP

When a surface condenser is fitted with an air ejector of the kinetic type the modern method of removing the condensate is by means of a centrifugal pump of special design, and known as a water-extraction pump or as a condensate pump. During the initial stages in the development of water-extraction pumps various difficulties and troubles arose, chief among these being the quiet and sudden failure of the pump to extract the water. The pump would run quite successfully and efficiently for a period, then it would suddenly and quietly fail and the condenser would begin to fill up; this resulted in a drop in vacuum owing to the rising water blanking off the ejector suction branch. The water was then drawn up into the ejector, which caused trouble.

This difficulty arose through air leakage at the pump glands; to get cool running at these glands the gland nuts had to be finger-tight, which resulted in air being drawn in, while, if the nuts were screwed up so as to prevent leakage, the shaft was scored or grooved and generally heated, and once this occurred it was very difficult to prevent leakage.

This air which leaked in entered the eye of the impeller and, since the air could not be discharged by the impeller owing to the difference in the density of air and water, the pump became virtually air-locked and its capacity consequently was reduced; the condenser therefore quietly filled up or flooded.

When it is appreciated that the air which leaks in immediately expands under the influence of the vacuum to about 1000 times its original volume, it may be realized that the minutest leakage is sufficient to cause this trouble.

Fig. 95 shows a section through the latest design of Messrs. The Mirrlees Watson Company for a horizontal water-extraction pump.

The pump is of the single-stage type, and twin impellers are employed working in parallel. The impellers are placed with the inlets facing each other, and have no balancing holes at the eyes. The impellers are shown at D, and are housed in a casing E, which is split horizontally to facilitate overhaul. Volutes B are arranged round the periphery of each impeller, and are suitably connected to a single discharge branch C. The condensate enters the common suction chamber A between the impellers, and each impeller deals with one-half the total condensate. Large clearances are arranged between the impellers and the casing, so that the discharge pressure is carried right to the stuffing glands, thus putting these under pressure. Any leakage, therefore, taking place at these glands is a leakage outwards of water and not a leakage inwards of air, and incidentally, no grease for the gland packing can contaminate the condensate.

A further feature of the design is the complete axial balance of the impellers, the end thrust of the one impeller being equal and opposite to the end thrust of the other impeller; this is due to the symmetrical arrangement of the impellers. The pump shaft is fitted with renewable gun-metal sleeves where the shaft passes through the stuffing boxes. These sleeves perform the double function of protecting the shaft and acting as distance pieces between the impellers and the double lock nuts which are provided, so that the impellers can be positioned centrally, in the pump casing. These pumps run normally at about 1500 r.p.m. for discharge heads from 5 to 50 ft. For higher heads or for lower speeds the impellers are arranged in series.

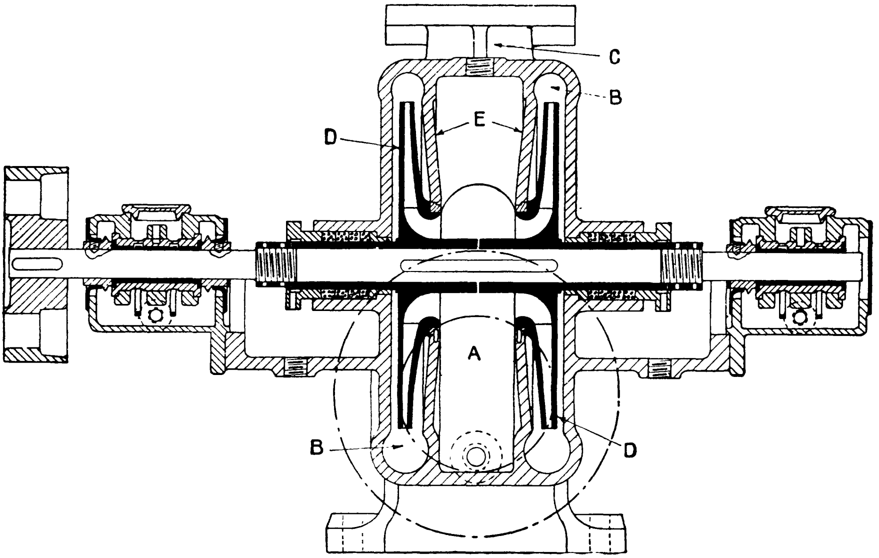


Fig. 95.—Mirrlees Extraction Pump

WEIR EXTRACTION PUMP

The water-extraction pump designed and manufactured by Messrs. G. & J. Weir, Ltd., is shown in fig. 96.

This pump is arranged with its shaft vertical, and employs twin impellers with a central suction. It is driven by a small steam turbine of the velocity-stage type, having three rows of moving blades mounted on a single wheel, and two rows of stationary blades mounted in a guide segment. A speed governor is fitted to maintain the speed constant under variations of load, and in addition a safety governor is fitted which shuts off the steam completely in the event of the pump overspeeding.

The twin impellers are shown at A, and B is the central suction chamber. The water delivered by the impellers is discharged through the diverging whirlpool chambers C to the discharge casing D on which the discharge branch is cast. No stuffing gland is employed at the lower impeller, and a large clearance is arranged between the top impeller and the cover, so that the full discharge pressure reaches the stuffing box E, thus preventing air leakage.

The weight of the impellers, turbine wheel shaft, &c., is carried on a ball thrust bearing F, and the shaft is centred by two ball journal bearings G and H. Both thrust and journal balls run in an oil bath formed by internal sleeves J and K, and a continuous oil circulation is maintained through the oil boxes L and M by the centrifugal action of the oil being rotated by the rotating balls and races. A normal design of carbon gland packing is employed for the turbine casing, and auxiliary nozzles under the control of auxiliary stop valves are provided, so that by regulating these auxiliary nozzles the pump may deal economically with large variations in load. The speed governor N is arranged at the top of the shaft and is of the centrifugal type; the variation in speed is 5 per cent from full load to the pump running empty. The overspeed governor is arranged directly beneath the speed governor and consists of an unbalanced ring held centrally by a spring, the whole rotating

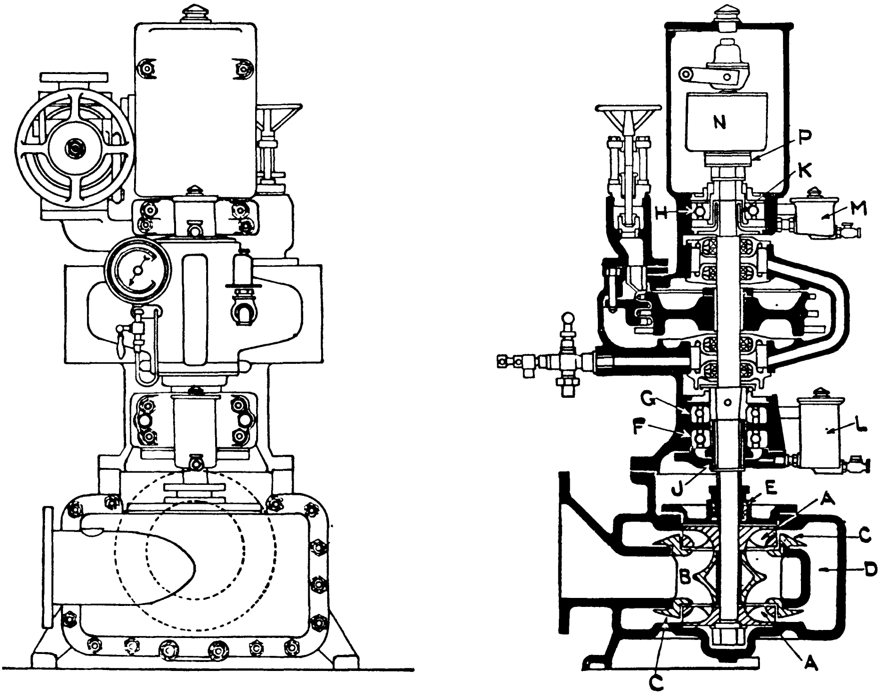


Fig. 96.—Weir Extraction Pump, Turbine Driven

with the shaft. At a predetermined speed the centrifugal force of the unbalanced ring overcomes the spring force and the ring flies out to its extreme range; the rotating ring then strikes a lever which disengages a trigger and allows a powerful spring attached to the stop valve to shut it, thus shutting off the steam to the pump.

The pump end is made of gun-metal, while the framing is of cast iron. The framing and pump end are divided down the centre line so that the shaft with impellers and turbine wheel can be removed for overhaul without disturbing any of the connections.

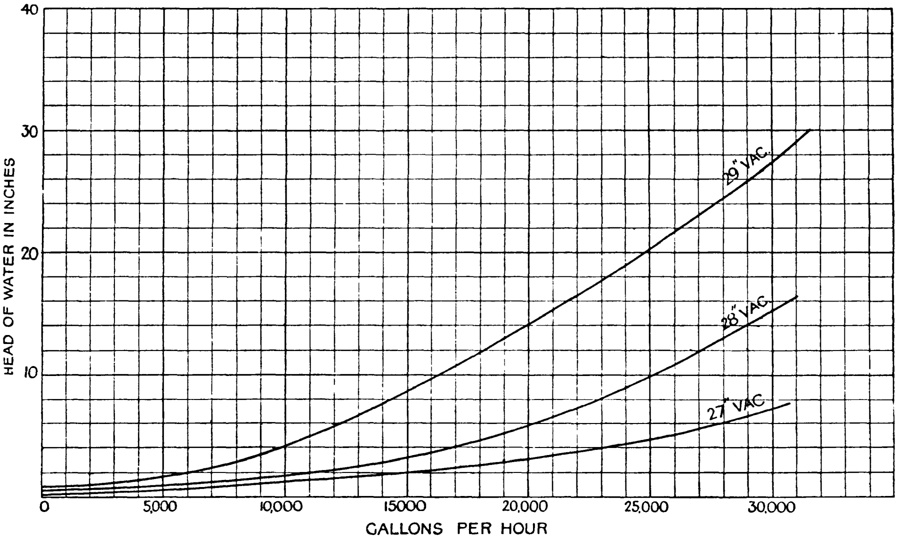


Fig. 97.—Weir Extraction Pump Suction Head Quantity Curves

The vertical design of the pump permits of the water end being placed at the lowest point. This is very important on board ship where the modern underhung condenser leaves little available head between the bottom of the condenser and the tank top.

Fig. 97 shows characteristic suction head quantity curves for these pumps.

DRYSDALE MOTOR-DRIVEN WATER-EXTRACTION PUMP

The latest design of water-extraction pump put on the market by Messrs. Drysdale & Co., Yoker, Glasgow, is shown in fig. 98. The pump is known as the " Pervac " patent pump, and the aim of the design is to satisfy the following fundamental conditions.

1. That air coming in through the suction pipe should be got rid of as promptly as possible.

2. That air leakage through joints, glands, &c., should be impossible.
3. That the suction branch should be as low as possible.
4. That the operation of the pump should be stable under all conditions of load from minimum to maximum.
5. That the efficiency should be as high as possible.
6. That the alignment of the pump and motor should not be disturbed by the expansion and contraction of the connecting pipes.

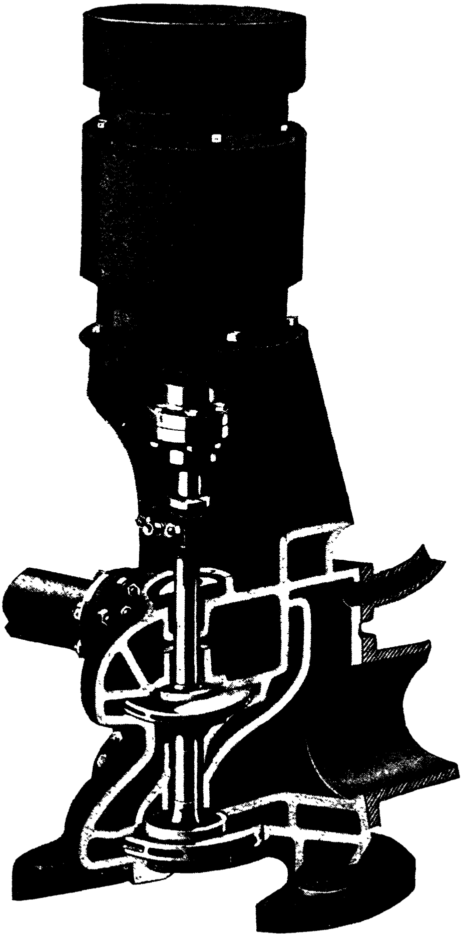


Fig. 98.—Drysdale "Pervac" Condensate Extraction Pump

The pump is of the two-stage vertical type with the driving motor mounted on the top on an extension of the main casting. By placing the pump vertical the suction branch can be placed low down. The pump casing is divided on the vertical plane and the suction and discharge branches are arranged on the back, thus permitting the pump to be dismantled without the breaking of pipe joints. The pump casing is of close-grained cast iron, while the impellers and packings are of gun-metal. The shaft is of nickel steel, and the coupling between the pump and motor is of the rigid pattern, fitted with adjustable plate. The internal bearings are of hard bronze of suitable composition to run with water lubrication only, while the external bearings are of the grease lubricated type, the arrangement being such that no grease can find its way into the condensate. The thrust bearing is incorporated in the top end shield of the motor.

Fig. 99 gives the pump characteristics for a constant speed. The actual quantity pressure curve is the line ABC. The drop in pressure from B to A is due to the friction head on the discharge side of the pump falling as the quantity is reduced. The dotted portion of the pressure curve BD does not actually exist, because the pump does not follow its head quantity curve, but fails in its theoretical performance owing to cavitation. The point B is the point of maximum output for the conditions. The horizontal line AE is the static discharge pressure on

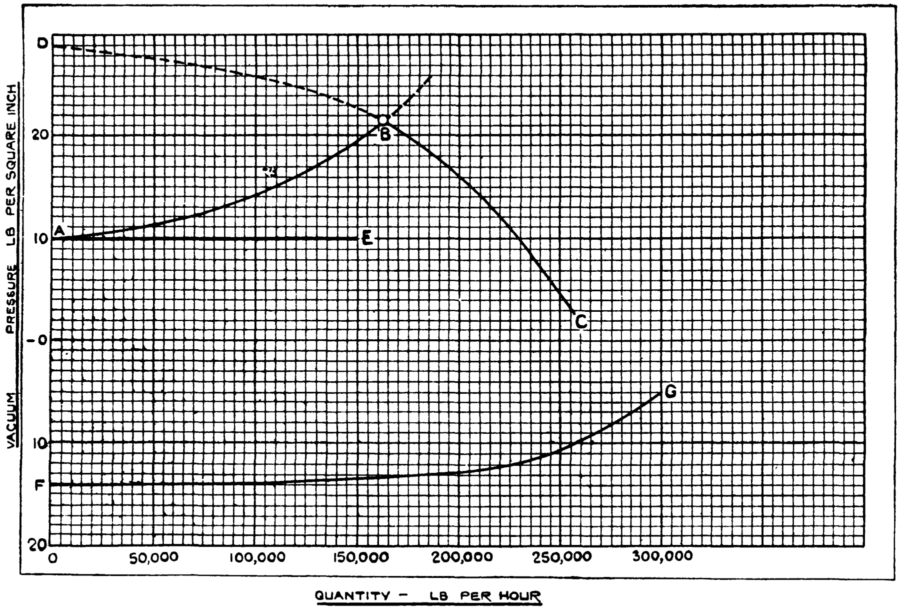


Fig. 99.—Characteristics for Drysdale Pump

the pump, and the curve FG shows the vacuum against which the pump can draw.

THEORY OF WATER-EXTRACTION PUMPS

Let V be the vacuum in the condenser in inches Hg;

h_s, the head of water in suction pipe over the centre of the impeller;

h, the total change in pressure on the water;

H, the discharge head in feet at pump discharge;

D, the outside diameter of the impeller in feet;

d, the inside diameter of the impeller in feet;

N, the revolutions per minute;

v_0, the peripheral velocity of the outside of the impeller in feet per second.

v_i, the peripheral velocity of the inside of the impeller vanes in feet per second;

v_{ri}, the radial velocity in feet per second at the entrance edges of the vanes;

v_{r0}, the radial velocity in feet per second at the leaving edges of the vanes;

θ_0, the vane angle at the leaving edge;

θ_i, the vane angle at the entrance edge;

V_{ri}, the relative velocity of water over vane at entrance;

V_{r0}, the relative velocity of water over vane at exit;

v_l, the actual velocity of water at exit; and

v_w, the velocity of whirl at exit.

The radial velocity v_{ri} is

$$v_{ri} = \sqrt{2gh_s}.$$

At the entrance edge we have the following velocity diagram (fig. 100).

$$\begin{aligned} \text{Then} \qquad V_{ri} &= v_{ri} \cos\theta_i \\ &= \sqrt{v_i^2 + v_{ri}^2}, \\ \text{and} \qquad v_i &= v_{ri} \cot\theta_i \\ &= \cot\theta_i \sqrt{2gh_s}. \end{aligned}$$

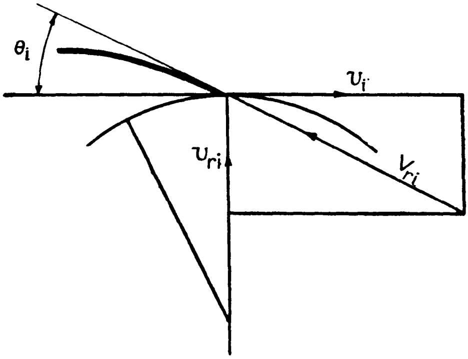


Fig. 100.—Velocity Diagram

Similarly at the exit edge we have the following velocity diagram (fig. 101).

$$\begin{aligned} \text{Then} \qquad v_{r0} &= (v_0 - v_w) \tan\theta_0, \ldots\ldots\ldots\ldots\ldots\ldots (74) \\ \text{and} \qquad V_{r0} &= v_{r0} \operatorname{cosec}\theta_0 \\ &= (v_0 - v_w) \tan\theta_0 \operatorname{cosec}\theta_0 \\ &= (v_0 - v_w) \sec\theta_0. \ldots\ldots\ldots\ldots\ldots\ldots (75) \end{aligned}$$

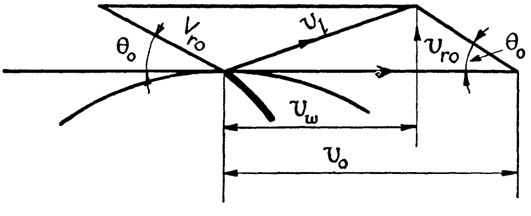


Fig. 101

If the impellers have a constant flow area, that is, if the area at entrance equals the area at exit then V_{ri} equals V_{r0}, and we get

$$v_{ri} \operatorname{cosec}\theta_i = (v_0 - v_w) \sec\theta_0.$$

The motion of a particle of water at any point in its passage through the impeller may be compounded into a velocity of whirl and one of flow parallel to the vanes. The change in pressure through the impeller will therefore be the sum of the changes due to the rotation of the water in a forced vortex and the flow parallel to the vanes.

For the forced vortex by considering only the entrance and exit edge of the vanes, and letting

H_i be the head at inlet due to forced vortex;
H_0, the head at outlet due to forced vortex;
w, the angular velocity of the wheel in radians per second;
r_i, the radius at inlet edges of vanes in feet; and
r_0, the radius at outlet edges of vanes in feet;

we have

$$\begin{aligned} H_i - w^2 - r_i^2 &= (H_0 - w^2 r_0^2)/(2g), \\ \text{or} \qquad H_0 - H_i &= (w^2 r_0^2 - w^2 r_i^2)/(2g). \end{aligned}$$

$H_0 - H_i$ is the change in head through the wheel due to the forced vortex;

$$\begin{aligned} \therefore \text{ change in head} &= (w^2 r_0^2 - w^2 r_i^2)/(2g) \\ &= (v_0^2 - v_i^2)/(2g). \end{aligned}$$

For the change in velocity of flow through the wheel we have, by Bernoulli's theorem,

$$\begin{aligned} \text{change of head} &= (V_{ri}^2 - V_{r0}^2)/(2g). \\ \text{Now} \qquad V_{ri} &= \sqrt{v_i^2 + v_{ri}^2}, \\ \text{and} \qquad V_{r0} &= v_{r0} \operatorname{cosec}\theta_0. \\ \therefore \text{ change in head} &= (v_i^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0)/(2g). \end{aligned}$$

The total change of head through the impeller will be the sum of these changes, or

$$\begin{aligned} \text{total change of head} &= (v_0^2 - v_i^2 + v_i^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0)/(2g) \\ &= (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0)/(2g) \ldots\ldots\ldots (76) \end{aligned}$$

If there be no change in the velocity of flow through the impeller, i.e. if $V_{ri} = V_{r0}$, then there is no change in head owing to change in the velocity of flow, and the whole gain in head is that due to the forced vortex motion. If V_{ri} is less than V_{r0}, which is generally the case, then the change in head due to change of flow is negative.

If there be no volute nor whirlpool chamber fitted to the pump, equation (76) will be the total gain in head, and the leaving velocity v_l will be entirely lost.

The total head to be overcome is made up of the discharge head at the pump discharge, the head over the impeller on the suction side and the head equivalent to the vacuum in the condenser, and the velocity head at the pump discharge.

The head equivalent to the vacuum V is (34V)/30, and the total change in head is

$$H - h_s + (34V)/30 + (v_d^2)/(2g),$$

where v_d is the discharge velocity.

$$\therefore H - h_s + (34V)/30 + (v_d^2)/(2g) = (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0)/(2g) \quad ..(77)$$

Like the ordinary centrifugal pump water-extraction pumps are usually fitted with a volute and a bell-mouthed discharge, or a volute and a whirlpool chamber. When a volute or whirlpool chamber is fitted to the pump, equation (77) has to be modified as follows.

If v_v be velocity of whirl in the volute casing, and assuming the radial velocity v_{r0} of the water leaving the impeller to be entirely dissipated in shock, then we have the water leaving the impeller with a velocity v_l, which velocity is changed to v_v in the volute casing.

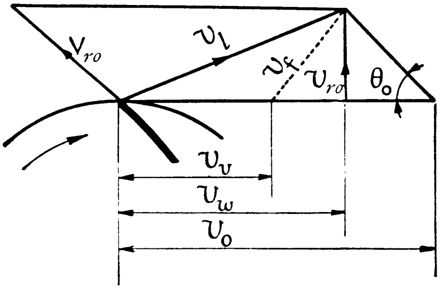


Fig. 102

Referring to fig. 102 the change of velocity is therefore v_f.

If the change be abrupt the loss of head is

$$\begin{aligned} v_f^2/(2g) &= \{(v_w - v_v)^2 + v_{r0}^2\}/(2g) \\ &= (v_w^2 - 2v_wv_v + v_v^2 + v_{r0}^2)/(2g) \\ &= (v_l^2 - 2v_wv_v + v_v^2)/(2g). \end{aligned}$$

If the change be gradual, then by Bernoulli's theorem the gain in head owing to the decreased velocity will be

$$(v_l^2 - v_v^2)/(2g).$$

The net gain in head will lie between the whole gain, due to the decreased velocity and the whole loss due to shock, and will be the difference between these two or

$$\text{the net gain in head} = (v_l^2 - v_v^2 - v_l^2 + 2v_wv_v - v_v^2)/(2g),$$

$$\text{i.e.} \quad H_g = 2(v_wv_v - v_v^2)/(2g).$$

The maximum possible gain may be found by differentiating this expression with respect to v_v and equating to zero, or

$$(dH_g)/(dv_v) = (v_w - 2v_v)/g.$$

Equating to zero we get
$$v_w - 2v_v = 0,$$
or
$$v_v = (v_w)/2.$$

That is the maximum gain in head, when the velocity of whirl in the volute is one-half the velocity of whirl of the water leaving the impeller.

Substituting this value of v_v in the equation for H_g we get the maximum gain to be

$$\begin{aligned} H_g &= [2\{\tfrac{1}{2}v_w v_w - (\tfrac{1}{2}v_w)^2\}]/(2g) \\ &= \{2(\tfrac{1}{2}v_w^2 - \tfrac{1}{4}v_w^2)\}/(2g) \\ &= \tfrac{1}{2}(v_w^2)/(2g). \quad \dots\dots\dots\dots \quad (78) \end{aligned}$$

That is, only one-half the available energy is gained, or the efficiency of the volute cannot be greater than 50 per cent. Actually the efficiency is very much below this, being generally about 10 to 15 per cent.

Adding this gain to equation (77) we get, for a pump fitted with a volute,

$$H - h_s + 34V/30 + v_d^2/(2g) = (v_v^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0 + \tfrac{1}{2}v_w^2)/(2g)\,..\,(79)$$

If a bell mouth be fitted to the discharge, the velocity v_v may be reduced and the discharge head increased correspondingly.

If v_p be the velocity in the discharge pipe, we have the gain of head

$$H_{pg} = (v_v^2 - v_p^2)/(2g).$$

If a whirlpool chamber be fitted round the impeller, it is usual to assume the motion through the chamber to be that of a free vortex.

Let r_1 be the inner radius of the vortex;
r_2, the outer radius of the vortex;
v_m, the velocity at the outer radius; and
v_l, the velocity at the inner radius.

Now for free vortex motion the gain in head will be

$$\begin{aligned} H_{vg} &= (v_l^2 - v_m^2)/(2g), \\ \text{also } v_l/v_m &= r_2/r_1 = \text{say } n. \\ \therefore H_{vg} &= \{v_l^2 - (v^2/n^2)\}/(2g) \\ &= \{v_l^2(1 - 1/n^2)\}/(2g). \\ \text{Again } v_l^2 &= v_w^2 + v_{r0}^2, \\ \therefore H_{vg} &= \{(v_w^2 + v_{r0}^2)(1 - 1/n^2)\}/(2g). \end{aligned}$$

Then adding this gain to equation (77) we get the head across the pump for a pump with a whirlpool chamber only to be

$$\begin{aligned} &H - h_s + 34V/30 + v_d^2/(2g) \\ &\quad = \{v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0 + (v_w^2 + v_{r0}^2)(1 + 1/n^2)\}/(2g)\,...\,(80) \end{aligned}$$

The actual gain in head given by a whirlpool chamber rarely exceeds 5 per cent of the total head, due to the modified form in which it is generally adopted. To get a large gain in head n must be large so that the dimensions become excessive; there is also the fact that in radial flow or divergent motion there is always the tendency to instability, which results in eddy losses which greatly reduce the efficiency. For this reason whirlpool chambers are sometimes made narrower at the outer diameter than at the inner diameter to reduce the decelleration of the water.

QUANTITY DISCHARGED BY PUMP

Let A be the peripheral area of the impeller at discharge in square feet, neglecting the thickness of the vanes this will equal $2\pi r_0 b_0$, where b_0 is the breadth;

Q, the quantity discharged in cubic feet per second.

The total gain in head is

$$H - h_s + 34V/30 + v_d^2/(2g) \text{ ft.}$$

The total rise of pressure on the water is

$$H - h_s + 34V/30 \text{ ft.} = h.$$

Then equation (77) becomes

$$h + v_d^2/(2g) = (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0)/(2g),$$

or

$$(2gh - v_d^2 - v_0^2 - v_{ri}^2)/\operatorname{cosec}^2\theta_0 = - v_{r0}^2.$$

If v_{ri} (the radial velocity at inlet) is equal to v_d the discharge velocity, the above expression becomes

$$\begin{aligned} v_{r0} &= \sin\theta_0\sqrt{v_0^2 - 2gh}, \\ \text{now } Q &= Av_{r0} \\ &= A \sin\theta_0\sqrt{v_0^2 - 2gh} \quad \ldots\ldots\ldots\ldots\ldots\ldots (81) \end{aligned}$$

HYDRAULIC EFFICIENCY OF PUMP

The hydraulic efficiency is the ratio

$$\frac{\text{total gain in pressure head across the pump}}{\text{head equivalent of the work done on pump spindle}}.$$

The total gain in pressure head across the pump is the discharge head H minus the suction head h_s plus the head equivalent to the vacuum $34V/30$ or $H - h_s + 30V/30$.

The head equivalent of the energy expended on the pump spindle may be found thus:

The change in angular momentum in foot-pounds per pound of water passing through the impeller

$$= (v_w - v_{wi})/g,$$

where v_{wi} is the velocity of whirl at the inlet edges of the vanes.

The change in the moment of this momentum will be

$$= (v_w r_0 - v_{wi} r_i)/g.$$

This change in the moment of the momentum is derived from the twisting moment T applied at the pump spindle.

$$\therefore \mathrm{T} = (v_w r_0 - v_{wi} r_i)/g.$$

Since the impeller is running at a speed of ω radians per second, the work done per second $= \{(v_w r_0 - v_{wi} r_i)\omega\}/g.$

Since this is the work done per second per pound of water, it is also equal to the head equivalent of the work done on the pump spindle.

Then, substituting these values, the hydraulic efficiency η is

$$\eta = g(\mathrm{H} - h_s + 34\mathrm{V}/30)/\{(v_w r_0 - v_{wi} r_i)\omega\}.$$

So far as the efficiency η is concerned the velocity of whirl v_{wi} must equal zero, since any whirl given to the water must have come from the impeller. Then, putting $v_{wi} = 0$, the expression reduces to

$$\eta = \{g(\mathrm{H} - h_s + 34\mathrm{V}/30)\}/(v_w r_0 \omega).$$

Now $v_0 = r_0 \omega,$

$$\therefore \eta = \{g(\mathrm{H} - h_s + 34\mathrm{V}/30)\}/v_w v_0),$$

or, since $v_w = v_0 - v_{r0} \cot\theta_0,$

$$\eta = \{g(\mathrm{H} - h_s + 34\mathrm{V}/30)\}/\{v_0(v_0 - v_{r0} \cot\theta_0)\} \ldots (82)$$

The power required to drive the pump will be

$$\mathrm{HP} = (v_w r_0 \omega \mathrm{WQ})/(550g); \ldots\ldots\ldots\ldots\ldots\ldots (83)$$

where W = the weight of 1 c. ft. of water in pounds.

The actual power required to drive the pump will be greater than that given by the above expression, because of the mechanical and hydraulic resistances.

SUCTION PIPES

Since the condenser is usually placed as low down as possible it is essential that the suction head over the eye of the impeller be reduced to a minimum. To attain this the loss of head along the suction pipe should be reduced to a minimum by keeping the velocity of flow low. The radial velocity of the water entering the eye of the impeller should be low.

A secondary reason why the suction pipe should be made large is to allow of any air which has leaked into the vacuum space of the pump, or the suction pipe itself, to flow back to the condenser and yet not restrict the flow to the pump sufficiently to cause erratic working of the pump.

The water velocity in the suction pipe should not exceed 3 ft. per second.

EXAMPLES

Example 1.—A single-stage water-extraction pump is to deal with 60,000 lb. of condensate per hour, when drawing from a 28 in. Hg vacuum and discharging against a static head of 5 ft. The head of water in the suction pipe must not exceed 3 ft. over the centre of the eye of the impeller. The bore of the discharge pipe is 3 in. Calculate the necessary diameter of the impeller at a speed of 800 r.p.m.

The exit angle $\theta_0 = 20^\circ$ and the discharge pipe is 30 ft. long, and a volute is fitted to the pump.

The friction loss will be

$$H_F = (4flv^2)/(2gm).$$

The friction loss in the suction pipe is here neglected, as the suction pipe velocity is usually made very low and the length of the pipe is kept as short as possible.

$$\begin{aligned} H_F &= (4 \times 0\cdot01 \times 30 \times 5\cdot46^2 \times 12)/(64\cdot4 \times 3) \\ &= 2\cdot21 \text{ ft.} \end{aligned}$$

The total change in head across the pump will be

$$\begin{aligned} H_t &= H - h_s + 34V/30 + v_d^2/(2g) \\ &= 5 + 2\cdot21 - 3 + (34 \times 28)/30 + 5\cdot46^2/64\cdot4 \\ &= 5 + 2\cdot21 - 3 + 31\cdot73 + 0\cdot462 \\ &= 36\cdot402 \text{ ft.} \end{aligned}$$

Taking the radial velocity $v_{ri} = v_{r0} = \sqrt{2gH_t}/4$ we get

$$\begin{aligned} v_{ri} = v_{r0} &= \sqrt{64\cdot4 \times 36\cdot402}/4 \\ &= 12\cdot1 \text{ ft. per second.} \end{aligned}$$

Again we have

$$\begin{aligned} v_{r0} &= (v_0 - v_w)\tan\theta_0, \\ \text{or} \qquad v_w &= -v_{r0}\cot\theta_0 + v_0 \\ &= v_0 - 12\cdot1 \times 2\cdot7475. \end{aligned}$$

Now from equation (79) we have

$$\begin{aligned} H - h_s + 34V/30 + v_d^2/(2g) &= (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0 + v_w^2/2)/(2g), \\ \text{or} \quad 36\cdot402 &= [v_0^2 + 12\cdot1^2 - 12\cdot1^2(1/0\cdot342)^2 + (v_0 - 33\cdot24)^2/2]/64\cdot4; \\ 2344 &= v_0^2 + 146\cdot4 - 146\cdot4 \times 8\cdot55 + (v_0^2 - 66\cdot48v_0 + 1104\cdot9)/2 \\ &= 1\cdot5v_0^2 - 33\cdot24v_0 - 552\cdot4; \\ v_0^2 - 22\cdot16v_0 &= 1791\cdot6; \\ (v_0 - 11\cdot08)^2 &= 11\cdot08^2 + 1791\cdot6; \\ v_0 &= 11\cdot08 \pm 42\cdot3 \\ &\doteq 53\cdot38 \text{ ft. per second (taking plus sign).} \end{aligned}$$

The revolutions per minute were given as 800, so that the necessary diameter of the impeller will be

$$
\begin{aligned}
800\pi D/(12 \times 60) &= 53{\cdot}38, \\
\text{or} \qquad D &= (53{\cdot}38 \times 12 \times 60)/(\pi 800) \\
&= 15{\cdot}29 \text{ in.}
\end{aligned}
$$

Example 2.—A water-extraction pump runs at 1000 r.p.m. and has an impeller 15 in. in diameter; the exit angle θ_0 of the vanes is 18°, and the static head on the discharge side is 10 ft., and on the suction side is 4 ft. The vacuum in the condenser is 29 in. Hg. The bore of the discharge pipe is 4 in. and the length 30 ft.; the breadth of the impeller at the periphery is $\frac{3}{4}$ in., and there are nine vanes each $\frac{1}{4}$ in. thick. Calculate the hydraulic efficiency and the horse-power, when the pump is dealing with 100,000 lb. per hour; neglect the friction loss in the suction pipe. The pump is single stage.

The total rise in pressure on the water across the pump is

$$H - h_s + 34V/30,$$

which equals

$$
\begin{aligned}
& 10 + 4flv^2/(2g) - 4 + (34 \times 29)/30 \\
=\ & 10 + (4 \times 0{\cdot}01 \times 30 \times 5{\cdot}11^2/64{\cdot}4) - 4 + (34 \times 29)/30 \\
=\ & 10 + 0{\cdot}48 - 4 + 32{\cdot}86 \\
=\ & 39{\cdot}34 \text{ ft.}
\end{aligned}
$$

The peripheral velocity v_0 is

$$
\begin{aligned}
v_0 &= (\pi \times 15 \times 1000)/(12 \times 60) \\
&= 65{\cdot}4 \text{ ft. per second.}
\end{aligned}
$$

To find the radial velocity v_{r0} we have

$$
\begin{aligned}
& \{(\pi \times 15 - 9/4)/12\}(0{\cdot}75/12)v_{r0} = 100{,}000/(3600 \times 62{\cdot}3), \\
& (3{\cdot}71 \times 0{\cdot}75/12)v_{r0} = 100{,}000/(3600 \times 62{\cdot}3), \\
& v_{r0} = (12 \times 100{,}000)/(3{\cdot}71 \times 0{\cdot}75 \times 3600 \times 62{\cdot}3) \\
& \quad\ = 1{\cdot}9 \text{ ft. per second.}
\end{aligned}
$$

The hydraulic efficiency η is obtained from equation (82)

$$
\begin{aligned}
\eta &= [g(H - h_s + 34V/30)]/[v_0(v_0 - v_{r0} \cot\theta_0)] \\
&= (32{\cdot}2 \times 39{\cdot}34)/[65{\cdot}4(65{\cdot}4 - 1{\cdot}9 \times 3{\cdot}077)] \\
&= 0{\cdot}3256 \\
\text{or} \qquad & 32{\cdot}56 \text{ per cent.}
\end{aligned}
$$

From equation (82) the horse-power necessary to drive the pump is

$$HP = (v_u r_0 \omega W Q)/(550g).$$

To find v_w we have from equation (74)

$$\begin{aligned} v_{r0} &= (v_0 - v_w)\tan\theta_0, \\ \text{or}\qquad v_w &= v_0 - v_{r0}\cot\theta_0 \\ &= 65{\cdot}4 - 1{\cdot}9 \times 3{\cdot}077 \\ &= 59{\cdot}56 \text{ ft. per second,} \\ \text{and}\qquad \omega &= 65{\cdot}4 \times 12/15 \\ &= 52{\cdot}3 \text{ radians per second.} \end{aligned}$$

$$\begin{aligned} \text{WQ} &= \text{the weight of water being dealt with in pounds per second} \\ &= 100{,}000/3600 \\ &= 27{\cdot}8 \text{ lb. per second.} \end{aligned}$$

Then substituting these numerical values in the HP equation we get

$$\begin{aligned} \text{HP} &= (59{\cdot}56 \times 15 \times 52{\cdot}3 \times 27{\cdot}8)/(32{\cdot}2 \times 12 \times 550) \\ &= 6{\cdot}1. \end{aligned}$$

The horse-power could also have been found by the external work done and the hydraulic efficiency, or

$$\begin{aligned} \text{HP} &= (100{,}000 \times 47{\cdot}38)/(33{,}000 \times 60 \times 0{\cdot}392) \\ &= 6{\cdot}1. \end{aligned}$$

Example 3.—In example 2 calculate what the maximum gain in head could be if a volute had been fitted to the pump, and calculate the hydraulic efficiency at the increased head.

Answer.—Gain in head (max.) = 27·52 ft.
Hydraulic efficiency = 56·14 per cent.

Example 4.—Find the relative velocity between the water and the vanes at exit of a water-extraction pump, having an impeller of 12 in. in diameter, running at 900 r.p.m. The radial velocity is 3·5 ft. per second, and the exit angle 10°. Find also the actual velocity of the water leaving the wheel.

Answer.—V_{r0} = 20·16 ft. per second.
v_l = 27·49 ft. per second.

Example 5.—If a whirlpool chamber be arranged round the impeller in example (4), and having an external diameter of 18 in., find the theoretical gain in head owing to the whirlpool chamber.

Answer.—6·516 ft.

Example 6.—Find the area for flow necessary at the periphery of the impeller of a water-extraction pump, which has to deliver 90,000 lb. ot condensate per hour against a discharge pressure of 5 lb. per square inch, and draw from a vacuum of 28 in. Hg, with a suction head over the eye of the impeller of 4 ft. The vane angle at the exit is 12°; the discharge pipe is 4 in. bore, the radial and peripheral velocities at inlet are 3 ft. per second and 55 ft. per second respectively.

Answer.—12·44 sq. in.

PRACTICAL CONSIDERATIONS WHICH AFFECT THE DESIGN OF WATER-EXTRACTION PUMPS

The foregoing theory of water-extraction pumps is based on the assumption that the pump runs full, and neglects (1) eddy losses, (2) impact losses, (3) unequal flow between vanes, (4) cavitation, &c.

For the theoretical calculations to coincide with actual tests coefficients of varying amount must be applied. These coefficients take the form of

$$2gH = Av_0^2 \pm Bv_0V_r \pm Cv_r^2.$$

The coefficients, A, B, and C vary very considerably in different pumps, and depend on the design, the shape, and the dimensions of the passages, &c.

One of the main conditions to be satisfied by a water-extraction pump is that it must lower the water level in the condenser down to the lowest limits to give good condenser drainage. This can only be accomplished by a centrifugal pump under conditions of cavitation, and it is the aim of all water-extraction pump designs to reduce the cavitation as much as possible. Since all cavitation cannot be avoided, it must be allowed for by making the inlet areas as large as possible. The large inlet area provided results in the eye of the impeller becoming abnormal, and the area through the impeller becomes a decreasing one. Bernoulli's change of head $(V_{ri}^2 - V_{r0}^2)/(2g)$ (see p. 188) therefore becomes negative.

The value of the radial velocity v_{ri} is usually only about 50 to 60 per cent of the available velocity, or

$$v_{ri} = 0{\cdot}5\sqrt{2gh_s}.$$

In general the inlet areas for a water-extraction pump are 2 to 3 times that given by the foregoing theory.

CHAPTER VIII

Circulating Pumps

The circulating pump used for circulating the cooling water in a modern surface condensing plant is invariably of the centrifugal type. These pumps are either motor driven or steam turbine driven.

The underlying principles governing the design of these pumps follow the generally accepted laws relating to centrifugal pumps. The formulæ deduced on pp. 187 to 192 for water-extraction pumps are applicable to these pumps, if the heads be modified to suit the new conditions under which the pump operates.

If H_p be the total suction and delivery head against which the pump operates, then equation (77), p. 189, becomes

$$H_p + v_d^2/(2g) = (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0)/(2g). \ldots\ldots(84)$$

If the pump be fitted with a volute, then equation (79) (p. 190) becomes

$$H_p + v_d^2/(2g) = (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0 + \tfrac{1}{2}v_w^2)/(2g). \ldots(85)$$

If a whirlpool chamber only be fitted, then equation (80), p. 190, becomes

$$H_p+v_d^2/(2g)=\{v_0^2+v_{ri}^2-v_{r0}^2 \operatorname{cosec}^2\theta_0+(v_w^2+v_{r0}^2)(1-1/n^2)\}/(2g). \ldots(86)$$

The quantity Q discharged by the pump will be the same as in equation (81), p. 191, since H_p is the total rise in pressure on the water, or the total head against which the pump delivers,

$$\text{i.e.} \qquad Q = A \sin\theta_0\sqrt{v_0^2 - 2gH_p}, \ldots\ldots\ldots\ldots\ldots\ldots(87)$$

and the power required to drive the pump will be

$$HP = (1/g)(v_w r_0 \omega)(WQ/550). \ldots\ldots\ldots\ldots\ldots(88)$$

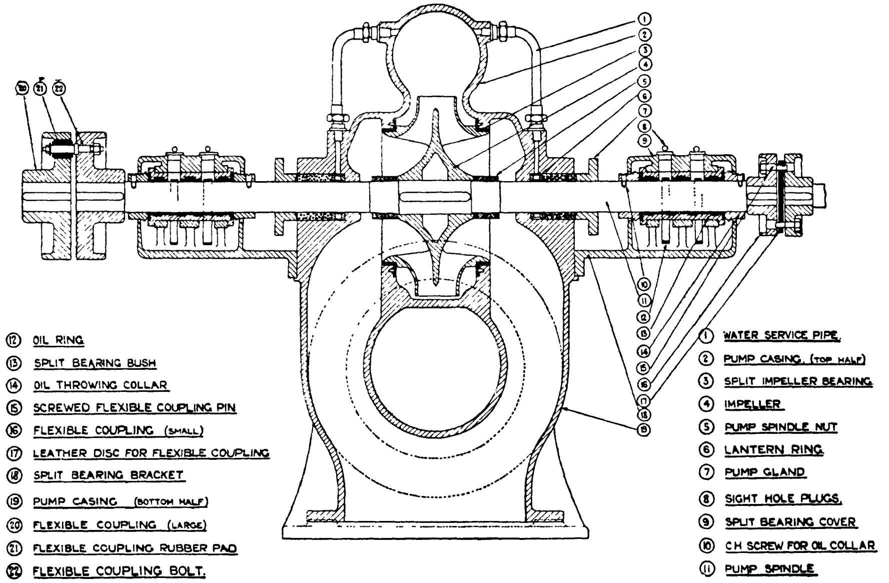
(See equation (83), p. 192.)

The total head H_p is made up of the friction head across the condenser (see pp. 17 to 19); the friction head in the suction and discharge piping, valves, &c., and, if the circulating water system is not siphonic, the static head.

The above formulæ do not quite agree with the results of experiment, and the reader should refer to a paper read by Dr. Blackadder (session 1923–1924) at the I.E.S., where the particular fact is brought out that the theoretical horse-power, as found from the above formula, is much in excess of the actual power taken by the pump. This, he remarks, is due to the water not leaving the pump at the dimensional angles, and to the pump not running equally full between the vanes.

DRYSDALE CIRCULATING PUMP

Fig. 103 shows a section through a modern centrifugal circulating pump as manufactured by Messrs. Drysdale & Co., Yoker, and as used for circulating a surface condenser. The impeller is of the double entry type which ensures axial balance. A plain volute is adopted with no whirlpool chamber. The impeller is keyed to the driving spindle and located by double nuts at each side, the whole being supported by external bearings lined with white metal. The bearings are ring oiled by the double oil rings (12). Collars (14) are fitted to each end of each bearing, so that the oil which escapes out of the bearings is thrown off and returns to the oil well. The pump stuffing boxes are fitted with lantern rings (6) and water service pipe (1) to water seal the stuffing boxes, and so prevent the entrance of air at these points which would upset the working of the pump. As normally installed, a centrifugal circulating pump has a suction or inlet casing pressure less than that of the atmosphere; a leakage at the pump glands would therefore be a leakage of air into the suction chamber unless means were adopted to prevent it, the pressure sealed stuffing box is therefore adopted.



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⑫ OIL RING
⑬ SPLIT BEARING BUSH
⑭ OIL THROWING COLLAR
⑮ SCREWED FLEXIBLE COUPLING PIN
⑯ FLEXIBLE COUPLING (SMALL)
⑰ LEATHER DISC FOR FLEXIBLE COUPLING
⑱ SPLIT BEARING BRACKET
⑲ PUMP CASING (BOTTOM HALF)
⑳ FLEXIBLE COUPLING (LARGE)
㉑ FLEXIBLE COUPLING RUBBER PAD
㉒ FLEXIBLE COUPLING BOLT.
① WATER SERVICE PIPE.
② PUMP CASING. (TOP HALF)
③ SPLIT IMPELLER BEARING
④ IMPELLER
⑤ PUMP SPINDLE NUT
⑥ LANTERN RING
⑦ PUMP GLAND
⑧ SIGHT HOLE PLUGS.
⑨ SPLIT BEARING COVER
⑩ C H SCREW FOR OIL COLLAR
⑪ PUMP SPINDLE

Fig. 103.—Drysdale Centrifugal Circulating Pump

This sub-atmospheric pressure in the suction or inlet casing is due to the normal location of the pump being above the suction water level, and is equal to the lift plus the reduction in pressure due to the friction loss in the suction pipe and the initiation of the flow along the pipe. Two types of flexible coupling are shown, the right-hand one consists of a disc of leather placed between the two faces of the coupling; pins (15) are screwed through each face of the coupling in alternate pairs; the pins have projecting ends which pass through the leather disc. The left-hand design incorporates rubber pads (21) located between the driving pins (22) and the half coupling (20). These flexible couplings are primarily intended to allow of end play in the armature of motor driven sets; they also allow of spring in the bedplate owing to the holding down bolts not being evenly tightened up.

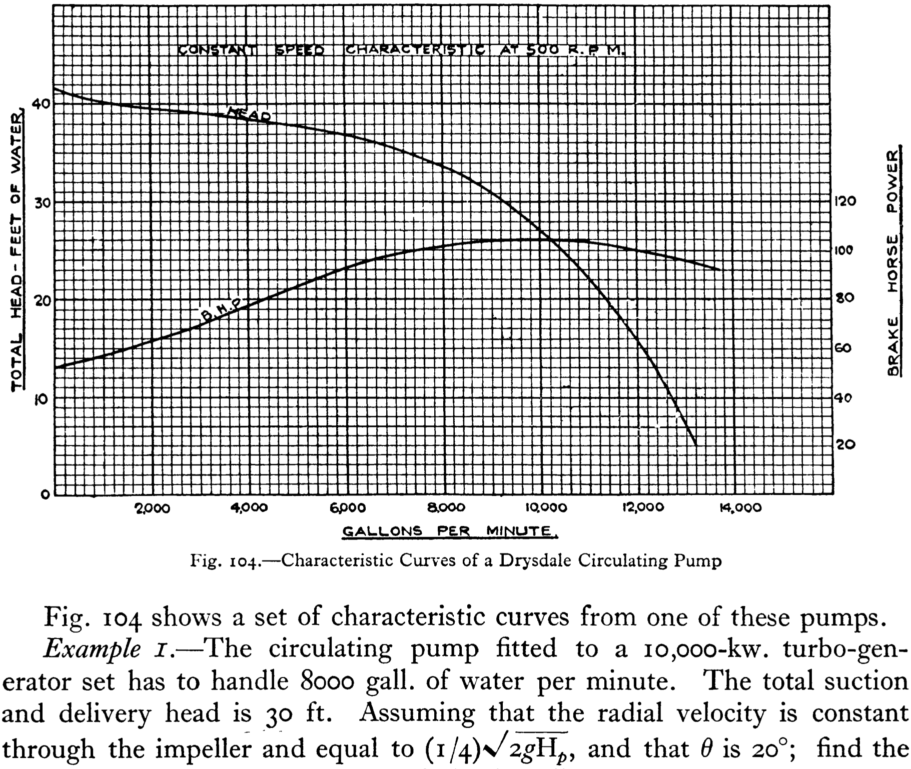

Fig. 104.—Characteristic Curves of a Drysdale Circulating Pump

Fig. 104 shows a set of characteristic curves from one of these pumps.

Example 1.—The circulating pump fitted to a 10,000-kw. turbo-generator set has to handle 8000 gall. of water per minute. The total suction and delivery head is 30 ft. Assuming that the radial velocity is constant through the impeller and equal to $(1/4)\sqrt{2gH_p}$, and that θ is 20°; find the necessary peripheral velocity v_0 of the impeller, and hence the necessary impeller diameter when the speed of the pump is 500 r.p.m. The pump is fitted with a plain volute, and the discharge velocity is 400 ft. per minute.

From equation (85), p. 197, we have,

$$H_p + v_d^2/(2g) = (v_0^2 + v_{ri}^2 - v_{r0}^2 \operatorname{cosec}^2\theta_0 + v_w^2/2)/(2g).$$

$$\text{Now} \quad v_d^2 = (400/60)^2$$

$$= 44{\cdot}4.$$

$$\text{And} \quad v_{ri}^2 = v_{r0}^2 = \{(1/4)\sqrt{2g \times 30}\}^2$$

$$= 64{\cdot}4 \times 30/16$$

$$= 120{\cdot}7.$$

$$\operatorname{cosec}^2\theta_0 = \operatorname{cosec}^2 20^c$$

$$= (1/0{\cdot}3420)^2$$

$$= 8{\cdot}55.$$

From equation (74), p. 187 we have

$$\begin{aligned} v_{r0} &= (v_0 - v_w)\tan\theta_0, \\ \text{or} \quad \sqrt{120\cdot7} &= (v_0 - v_w)\tan 20^\circ, \\ \text{or} \quad v_w &= v_0 - 10\cdot98/0\cdot364 \\ &= v_0 - 30\cdot16. \end{aligned}$$

Then, substituting these numerical values in the above equation, we get

$$30 + 44\cdot4/64\cdot4 = \{v_0^2 + 120\cdot7 - 120\cdot7 \times 8\cdot55 + (v_0 - 30\cdot16)^2/2\}/64\cdot4,$$

i.e. $v_0^2 - 20\cdot12 v_0 = 1621\cdot2$.

$$\therefore\ v_0 = 51\cdot56 \text{ ft. per second.}$$

If D be the impeller diameter in inches,

$$\begin{aligned} \text{then} \quad (\pi \times D \times 500)/(12 \times 60) &= 51\cdot56, \\ \text{or} \quad D &= (51\cdot56 \times 12 \times 60)/(\pi \times 500) \\ &= 23\cdot62 \text{ in.} \end{aligned}$$

Example 2.—Calculate the hydraulic efficiency η and the horse-power necessary to drive the pump given in example 1.

From equation (82), p. 192, we have

$$\eta = g\{H - h_s + (34V/30)\}/\{v_0(v_0 - v_{r0}\cot\theta_0)\}.$$

The quantity $\{H - h_s + (34V/30)\}$ is the total pressure rise which in a circulating pump is h, and the above expression therefore becomes

$$\eta = gh/\{v_0(v_0 - v_{r0}\cot\theta_0)\}.$$

Now from example 1,

$$\begin{aligned} h &= 30 \text{ ft,} \\ v_0 &= 51\cdot56 \text{ ft. per second,} \\ v_{r0} &= \sqrt{120\cdot7}, \\ \cot\theta_0 &= \cot 20^\circ = 2\cdot7475. \end{aligned}$$

Then, substituting these numerical values, we get

$$\begin{aligned} \eta &= 32\cdot2 \times 30/\{51\cdot56(51\cdot56 - 2\cdot7475\sqrt{120\cdot7})\} \\ &= 32\cdot2 \times 30/(51\cdot56 \times 21\cdot4) \\ &= 0\cdot87. \end{aligned}$$

From equation (88), p. 197, the horse-power necessary to drive the pump is

$$HP = (1/g)(v_w r_0 \omega)(WQ/550).$$

From example 1 we get:

$$\begin{aligned} v_w &= v_0 - 30\cdot16 \\ &= 51\cdot56 - 30\cdot16 \\ &= 21\cdot4 \text{ ft. per second,} \\ r_0 &= 23\cdot62/(2 \times 12) \\ &= 0\cdot984 \text{ ft.} \\ \omega &= 2\pi \times 500/60 \\ &= 52\cdot4 \text{ radians per second,} \\ Q &= 80{,}000/(60 \times 62\cdot3) \\ &= 21\cdot4 \text{ c. ft. per second,} \\ W &= 62\cdot3 \text{ lb. per cubic foot.} \end{aligned}$$

Then, substituting these numerical values, we get

$$\begin{aligned} \text{HP} &= (1/32\cdot2)(21\cdot4 \times 0\cdot984 \times 52\cdot4)(62\cdot3 \times 21\cdot4)/550 \\ &= 83. \end{aligned}$$

Example 3.—During the test of a surface condensing plant the following observations were made.

Quantity of steam condensed in pounds per hour, 83,260.
Pressure gauge reading on centrifugal pump suction, 5 in. Hg vacuum.
Pressure gauge reading on centrifugal pump discharge, 20 ft. head.
Temperature of inlet water to main condenser, 50° F.
Temperature of outlet water from main condenser, 66° F.
Voltage of supply current to centrifugal pump motor, 400.
Current, D.C.
Reading of ampere meter on pump motor, 208.

Assuming that each pound of steam condensed in the condenser gave up 990 B.Th.U. to the circulating water, estimate the efficiency of the circulating pump if the motor efficiency is 80 per cent.

From equation (16), p. 16, the quantity of circulating water will be

$$G = W(qL + t_1 - t_2)/\{600(T_2 - T_1)\}$$

The quantity $(qL + t_1 - t_2)$ is the heat imparted to the circulating water, and therefore is 990.

$$\begin{aligned} \text{Then } G &= 83260 \times 900/\{600\,(66 - 50)\} \\ &= 8590 \text{ gall. per minute} \\ &= 85{,}900 \text{ lb. per minute.} \end{aligned}$$

The total head across the pump

$$\begin{aligned} &= \text{head equivalent to vacuum on suction side} + \text{discharge head} \\ &= (5 \times 34/30) + 20 \\ &= 5\cdot67 + 20 \\ &= 25\cdot67 \text{ ft.} \end{aligned}$$

The pump water horse-power will therefore be

$$\begin{aligned} \text{WHP} &= 85{,}900 \times 25{\cdot}67/33{,}000 \\ &= 66{\cdot}8. \end{aligned}$$

The electrical horse-power supplied to the motor is

$$\begin{aligned} \text{EHP} &= \text{volts} \times \text{amperes}/746 \\ &= 400 \times 208/746 \\ &= 111{\cdot}7. \end{aligned}$$

Since the motor efficiency is 80 per cent, the brake horse-power absorbed at the motor spindle will be

$$\begin{aligned} \text{BHP} &= 111{\cdot}7 \times 0{\cdot}8 \\ &= 89{\cdot}4. \end{aligned}$$

The pump efficiency will therefore be

$$\text{WHP}/\text{BHP} = 66{\cdot}8/89{\cdot}4 = 0{\cdot}746.$$

SUCTION AND DISCHARGE PIPING FOR CIRCULATING WATER PUMPS

The lay out of the circulating or cooling water piping in a modern land surface condensing plant is arranged in either of two ways, according to whether cooling towers are employed or not. When cooling towers are used the discharge is open and the circulating pump must overcome the static head. If the circulating water be drawn from a river or from the sea, or from where there is an abundant supply of cold water, the suction and discharge piping is usually arranged so that the flow is siphonic. By this means the total head to be overcome by the circulating pump is the hydraulic friction in the piping, valves and condenser only. The siphonic arrangement thus affects a considerable saving in the power required to drive the circulating pump, and this arrangement is always adopted where practicable.

The circulating pump should be located in the pipe line, so that the length of suction piping is as short as possible, and the condenser should be on the discharge side of the pump. Centrifugal pumps always operate more stably when the greatest proportion of the total head is on the discharge side, and for this reason the suction head should be reduced to a minimum by keeping the piping as short and with as few bends as possible. The suction lift or the height at which the pump is placed above the water level in the river should also be kept as small as possible. To satisfy this condition it is common practice in power stations to have a separate pump house placed close to the river from which the pump draws, and in some cases vertical spindle pumps have been adopted, so that the pump could be placed very low down.

The calculation of the pipe friction follows the same general law as was given on p. 82, that is

$$H_p = flv^2/(2gm),$$

for a straight pipe of length l ft., and allowing as before 15 diameters for every easy bend, the above formula becomes

$$H_p \{ fv^2(l + 15Dn) \} /(2gm).$$

For circular pipes this becomes, since $m = D/4$,

$$H_p = \{ 4fv^2(l + 15Dn) \} /(2gD) \ldots\ldots\ldots\ldots\ldots (89)$$

The numerical value of the constant f was given on p. 82 as 0·01, this constant, however, varies with the velocity v and with the diameter D. It also varies with the roughness of the surface, and is less for new pipes than for old ones. f diminishes with increase in velocity and as the diameter D increases. For cast-iron pipes from 10 to 15 in. in diameter, f may be taken at 0·01; and from 10 to 5 in., 0·012; and from 5 in. downward, 0·014.

For practical computations of the probable friction loss in any circulating water-pipe line system the variation in f with velocity may be neglected, as the above values of f are on the safe side, unless for very old pipe lines which are badly encrusted. To the above loss must be added the losses due to valves, sudden enlargements, strainers, &c., which may be in the line. To reduce these losses to the minimum, full-way valves are always employed, and sudden enlargements can be avoided by the fitting of taper distance pieces where necessary.

The loss due to a sluice valve may be found from the following formula by Weisbach.

$$\text{The loss } H_v = Fv^2/(2g), \ldots\ldots\ldots\ldots\ldots\ldots (90)$$

where F has the following values according to the ratio r of the distance the valve is open to the diameter of the valve.

r	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1·0
F	98	17	5·5	2·0	0·8	0·25	0·07	0·0

Example 1.—The circulating water system on a land surface condensing plant is arranged siphonically. The total length of the suction and discharge piping is 150 ft., the quantity of water being pumped by the circulating pump is 12,000 gall. per minute, and the bore of the piping is 24 in. There are nine easy right-angled bends in the line; one sluice valve is fitted to the suction of the pump, and one to the discharge. The friction head in the condenser is 15 ft., and a sluice valve is fitted to the outlet branch. Estimate the total head against which the circulating pump must operate when the sluice valve on the condenser outlet is three-quarters open.

From equation (89) the pipe friction will be

$$H_p = \{4fv^2/(2gD)\}(l + n15D).$$

$$\text{Now} \quad v = (120{,}000 \times 144)/(62{\cdot}3 \times 452{\cdot}4 \times 60)$$
$$= 10{\cdot}22 \text{ ft. per second.}$$

f may be taken at 0·01.

$$\text{Then } H_p = \{4 \times 0{\cdot}01 \times 10{\cdot}22^2/(64{\cdot}4 \times 2)\}(150 + 9 \times 15 \times 2)$$
$$= 13{\cdot}62 \text{ ft.}$$

The loss at the outlet sluice valve is found from equation (90):

$$H_v = Fv^2/(2g)$$
$$= 0{\cdot}25 \times 10{\cdot}22^2/64{\cdot}4$$
$$= 0{\cdot}405 \text{ ft.}$$

Since the discharge velocity will all be lost, the pump must maintain the head equivalent to this velocity, which is

$$H_L = v^2/(2g)$$
$$= 10{\cdot}22^2/64{\cdot}4$$
$$= 1{\cdot}62 \text{ ft.}$$

The total head against which the pump must discharge is therefore

$$H = H_p + H_v + H_L + 15$$
$$= 13{\cdot}62 + 0{\cdot}405 + 1{\cdot}62 + 15$$
$$= 30{\cdot}645 \text{ ft.}$$

Example 2.—The condensing plant on board a merchant ship is fitted with a turbine-driven circulating pump, having a steam consumption rate of 50 lb. per WHP hour. The friction head across the condenser is 21 ft., and when the ship is light the vertical distance between the centre line of the discharge overboard and the sea level is 20 ft. The total quantity of water circulating through the main condenser is 9000 gall. per minute, and the circulating water pipes are 20 in. in diameter, and have a total length of 40 ft. with 6 easy bends in the line. The loss in head across the inlet and outlet ship's side valves and gratings is 6 ft. Estimate the total head against which the pump operates and the quantity of steam used by the pumps.

$$\text{The velocity } v = 90{,}000 \times 144/(62{\cdot}3 \times 60 \times 314{\cdot}16)$$
$$= 11 \text{ ft. per second.}$$

The pipe friction H_p will be

$$H_p = \{4 \times 0{\cdot}01 \times 11^2/(64{\cdot}4 \times 1{\cdot}669)\}(40 + 6 \times 15 \times 1{\cdot}669)$$
$$= 8{\cdot}56 \text{ ft.}$$

The head equivalent to the discharge velocity is

$$H_L = v^2/(2g)$$
$$= 11^2/64{\cdot}4$$
$$= 1{\cdot}88 \text{ ft.}$$

The total head against which the pump operates is $H_p + H_L +$ static head above the sea level + the loss across the ship's side valves + condenser friction

$$= 8{\cdot}56 + 1{\cdot}88 + 20 + 6 + 21$$
$$= 57{\cdot}44 \text{ ft.}$$

The

$$\text{WHP} = 90{,}000 \times 57{\cdot}44/33{,}000$$
$$= 156{\cdot}6;$$

and the steam consumption will be

$$= 50 \times 156{\cdot}6$$
$$= 7833 \text{ lb. per hour.}$$

Example 3.—A land surface condensing plant has cooling towers provided to cool the cooling water. The vertical distance between the water level in the suction culvert and the discharge pipe entering the top of the cooler is 40 ft. There is a non-return, multiple-shutter valve fitted to the discharge branch of the circulating pump, and the friction head across this valve is 3 ft. The total length of suction and discharge piping is 200 ft. and the bore 26 in., and there are 12 easy right-angled bends in the piping. The friction head in the condenser is 16 ft. Calculate the total head against which the pump works when 15,000 gall. of water are being pumped per minute, and calculate the brake horse-power necessary at the pump spindle if the pump efficiency is 75 per cent.

The water velocity v in the piping is

$$v = 150{,}000 \times 144/(62{\cdot}3 \times 530{\cdot}9 \times 60)$$
$$= 10{\cdot}9 \text{ ft. per second.}$$

The pipe friction will be

$$H_p = \{4 \times 0{\cdot}01 \times 10{\cdot}9^2/(2 \times 32{\cdot}2 \times 2{\cdot}165)\}(200 + 12 \times 2{\cdot}165 \times 15)$$
$$= 19{\cdot}8 \text{ ft.}$$

The head equivalent to the discharge velocity

$$H_L = v^2/(2g)$$
$$= 10{\cdot}9^2/64{\cdot}4$$
$$= 1{\cdot}844 \text{ ft.}$$

The total head against which the pump works is

$$= 19{\cdot}8 + 1{\cdot}844 + \text{static head} + \text{condenser friction}$$
$$= 19{\cdot}8 + 1{\cdot}844 + 40 + 16$$
$$= 77{\cdot}6 \text{ ft.}$$

The brake horse-power at the pump spindle will be

$$\text{BHP} = 150{,}000 \times 77{\cdot}6/(33{,}000 \times 0{\cdot}75)$$
$$= 470.$$

CHAPTER IX

Evaporators

It is rapidly becoming standard practice to instal evaporators in power stations to supply the necessary make-up water. The evaporator supercedes the water softening plant, and in the modern installation having water tube boilers and steam turbines, it has many advantages over the water softening plant.

The majority of natural waters are hard, that is, they have the power of destroying soap solution. The hardness is caused by the bicarbonates or sulphates of lime and magnesia. When bicarbonates are present the hardness is said to be " temporary ", and hardness due to sulphates is said to be " permanent "; both, however, may be present in the same water. If water containing bicarbonates be boiled, these are decomposed with precipitation of most of the lime and magnesia. If the water contains sulphates, these are not removed by boiling; by suitable chemical means, however, they may be removed more or less completely, but the total amount of solids in the water is not affected, because the water after treatment contains an amount of sodium salts equivalent to the lime and magnesia salts which were present before treatment. The concentration of these solids takes place in the boiler, and they must be removed by the blowing down of the boiler. This involves a serious loss of heat. The principal advantage of the softening process is the prevention of the rapid scaling of the heating surfaces of the boiler due to the insoluble lime and magnesia salts.

The evaporator produces pure distilled water free from solids, so that the loss of heat due to boiler blow down is eliminated. Owing to the purity of the water and the freedom from scale, the boilers can be worked at a much higher steaming rate, and their efficiency thereby improved. These facts were recognized by the British Admiralty many years ago, and they laid down the rule that the initial filling of naval boilers was to be done with pure distilled water and that all make-up water used thereafter was to be pure distilled water. It was only by this means that the high rate of steaming demanded from these boilers could be obtained and maintained without priming.

On land power stations the use of distilled water allows the peak loads to be comfortably met owing to the increased steaming capacity obtained; it is claimed that the emergency boiler capacity obtained is nearly doubled by the use of distilled water and that without fear of priming. Owing to the elimination of the blow down the necessary quantity of make-up required is reduced from about 5 to 6 per cent to 3 to 4 per cent. Since the raw water fed to the evaporator may be drawn from any source, it can frequently be obtained without charge, whereas the water fed to a water softening plant should be the best obtainable and is generally purchased at considerable expense. If the evaporator be installed so that the heat in its vapour is utilized in the general feed heating scheme, the operating charges become very small; approximately 95 per cent of the total heat supplied to the evaporator can be recovered in the feed water.

It is usual to instal single effect evaporators; but double or triple effect may be installed, if the efficiency of the feed heating scheme demand it.

THE WEIR EVAPORATOR

Figs. 105 and 106 show two views of a Weir vertical type evaporator as installed in land power stations. The heating surface is made up of heavy gauge copper tube coils (11) coiled in a horizontal plane; the steam passes through the interior of these coils, while the water being evaporated is on the outside; each coil is independent, and may be removed for cleaning quite independent of the others. A steam pocket (10) and a drain pocket (8) are cast integral with the shell of the evaporator, and the ends of the coils are fixed to these pockets by special gun-metal couplings, one end of the coil being fixed to the steam pocket and the other end to the drain pocket. The full area of the tube is arranged in the coupling at the inlet end, but

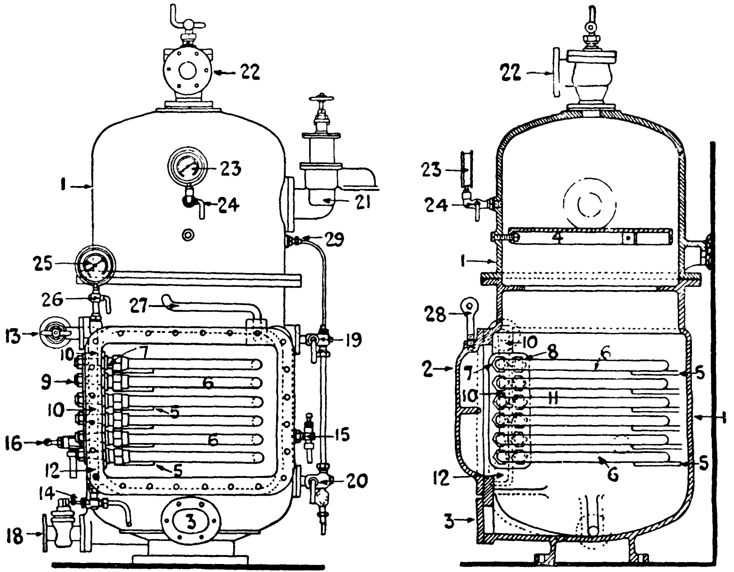

Fig. 105 Fig. 106

The Weir Evaporator

the area is restricted at the outlet or drain end. The bottom coil is arranged as a drain coil, that is, the drains from all the upper coils are collected at the bottom of the drain pocket and returned through the bottom coil to a drain chamber partitioned off from the steam pocket. The drain outlet is at (12). This arrangement of coils ensures a complete condensation of the steam without the necessity for continual regulation of the drain valve (14). To the steam inlet branch (10) is fitted the steam inlet valve (13) and pressure gauge (25). The shell of the evaporator is of cast iron and is fitted with safety valve (21), pressure gauge (23), vapour outlet valve (22), water-level gauge cocks (19 and 20), blow-off cock (18), and automatic water-level regulator. A large door (2) is fitted to the front of the evaporator, so that the coils can be easily removed for cleaning purposes, the easy removal of the door is facilitated by the crane bar (27).

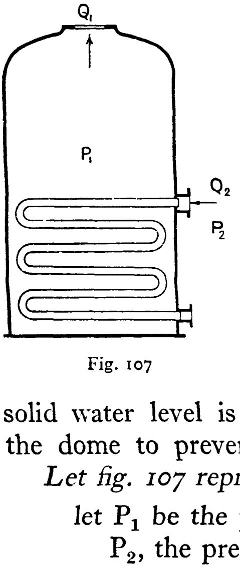

Fig. 107

The water level in these evaporators is normally arranged so that only about one-half of the coils are immersed by solid water, as it has been found by experience that good water is produced by carrying this water level without any reduction in quantity owing to the upper coils being out of solid water. This is due to the violent foaming and ebullition which takes place causing the water level to virtually rise, so that the upper coils are really evaporating or drying the steam. This has given to the upper coils the name of drying coils. The top of the water-level gauge glass is connected by the pipe (29) well up in the steam or vapour space, so that a solid water level is shown in the glass. A baffle (4) is arranged inside the dome to prevent priming.

Let fig. 107 represent an evaporator, and

let P_1 be the pressure in the shell;
P_2, the pressure in the coils;
V_1, the volume of the vapour at pressure P_1;
V_2, the volume of the steam at pressure P_2;
S, the surface in square feet in the coils;
t_1, the steam temperature at pressure P_1;
t_2, the saturation temperature at pressure P_2,
t_f, the temperature of the feed water entering;
t_d, the temperature of the drain;
L_1, the latent heat of the vapour at pressure P_1;
L_2, the latent heat of the steam at pressure P_2;
Q_1, the quantity of vapour produced in pounds per hour; and
Q_2, the quantity of steam used in pounds per hour.

The heat given up by the steam in condensing in the coils and being cooled to temperature t_d

$$= Q_2(L_2 + t_2 - t_d) \text{ B.Th.U. per hour;}$$

or, if the dryness fraction of the entering steam be q,

$$= Q_2(qL_2 + t_2 - t_d).$$

The heat absorbed by the vapour in being produced from water at t_f° F.

$$= Q_1(L_1 + t_1 - t_f) \text{ B.Th.U. per hour.}$$

Neglecting radiation these two quantities of heat must be equal.

$$\therefore\ Q_2(qL_2 + t_2 - t_d) = Q_1(L_1 + t_1 - t_f) \ldots\ldots\ldots (91)$$

With a properly clad evaporator the radiation losses seldom exceed 2 per cent, so that, allowing this value for the radiation loss, the above equation becomes

$$0{\cdot}98\{Q_2(qL_2 + t_2 - t_d)\} = Q_1(L_1 + t_1 - t_f) \ldots\ldots\ldots (92)$$

If the steam supply to the evaporator be superheated and if the temperature of the superheated steam be t_s, then the heat given up by the steam is

$$Q_2\{L_2 + \gamma(t_s - t_2) + t_2 - t_d\},$$

where γ is the specific heat of the steam at pressure P_1. The numerical value of γ may be found from one of the published diagrams, such as that given in *Steam Turbines* by Goudie, or the heat given up by the steam may be found from the difference in total heats at pressure P_1 and temperature t_s, and pressure P_1 and saturation temperature t_2, from the steam tables.

If H_2 be the total heat at temperature t_2 and pressure P_2; and H_s the total heat at temperature t_s and pressure P_2, then the heat given up by the steam is

$$Q_2\{(H_s - H_2) + L_2 + t_2 - t_d\};$$

and equation (91) becomes

$$0{\cdot}98Q_2\{(H_s - H_2) + L_2 + t_2 - t_d\} = Q_1\{L_1 + t_1 - t_f\} \ldots (93)$$

From these expressions the quantity of vapour produced by the evaporator may be found for given conditions of supply and production.

SURFACE IN EVAPORATORS

Owing to the indeterminate and varying water velocity which takes place inside an evaporator shell, it is impossible to make allowance for the variation in heat flow with velocity, as was done in our consideration of surface condensers. An allowance may be made for the viscosity effect of the cold feed water entering the evaporator, but since the effect penetrates so short a distance into the evaporator and since the water is evaporated as well as heated, the viscosity effect may be neglected. The proportion of heat absorbed by the water during the period it is being heated is small compared with the proportion absorbed during evaporation.

When fixing up the surface necessary in an evaporator, it is customary, therefore, to allow a heat flow of so many B.Th.U. per square foot of surface per hour per degree F. of mean temperature difference. This heat flow is known as the rate of transmission of heat or transmission rate. The actual value allowed should vary with the shell pressure. This is due to the presence of air in the water fed into the evaporator. This air is liberated mainly on the tube surface, and the lower the shell pressure the greater will be the volume of air liberated. The increased air volume at reduced shell pressures will blanket off an increased proportion of the tube surface, so that the allowable transmission rate will progressively decrease as the shell pressure decreases. This appears to be borne out from test results.

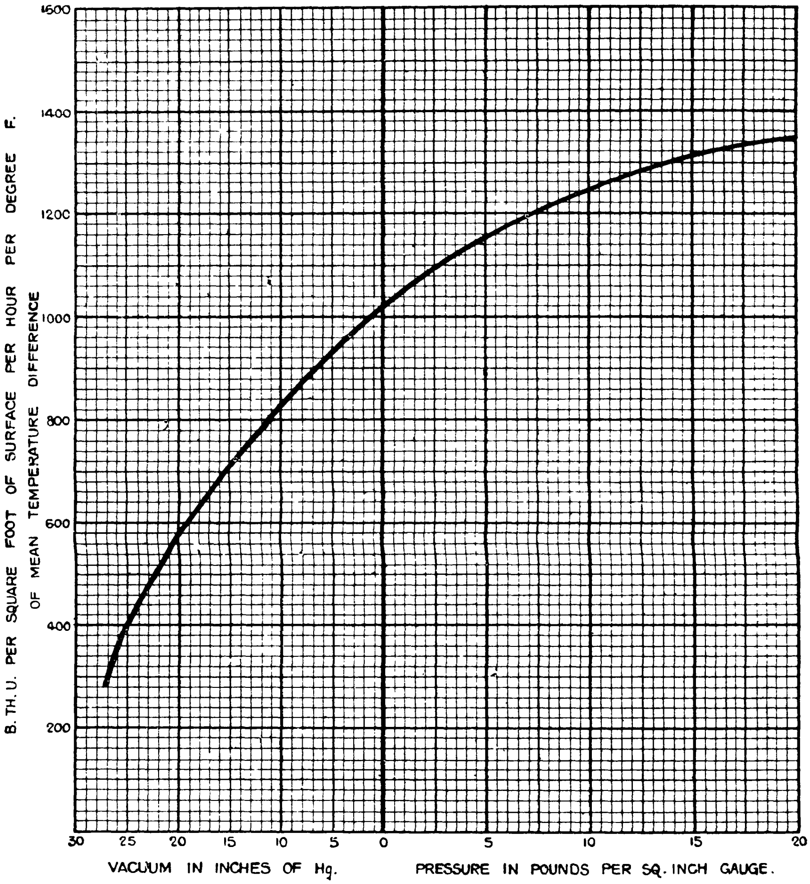

Fig. 108.—Transmission Rates in Evaporators with Copper Coils

Fig. 108 gives working values of the transmission rate for various shell pressures for copper tubes, which have been found by the author to give reasonable results; for iron or steel tubes these should be reduced 40 per cent. These values are about the maximum which can be attained in practice, and they do not include a working margin to allow for the scaling of the coils.

If θ_m is the mean temperature difference between the steam and the water;

R, the rate of transmission of heat from the steam to the water; and

H_F, the heat flow from the steam to the water;

we have

$$S = H_F/(\theta_m R). \qquad\qquad (94)$$

The heat flow H_F equals the heat absorbed by the vapour being produced from water at t_f° F.

$$\therefore S = Q_1(L_1 + t_1 - t)/(\theta_m R). \qquad\qquad (95)$$

The heat flow also equals the heat given up by the steam in condensing under the specific conditions.

MEAN TEMPERATURE DIFFERENCE

The true mean temperature difference in an evaporator cannot be found by Grashof's equation, as given on p. 4, because the heating of the feed water entering the evaporator is carried on to the evaporation of the water.

If the feed water enters the evaporator at the temperature of evaporation and the steam is just condensed, that is, the drain from the evaporator coils is at the steam temperature, then the mean temperature difference is the steam temperature minus the evaporation temperature. This may be shown, fig. 109, by taking one coil and straightening it out and plotting a diagram of the temperature which obtains along the coil.

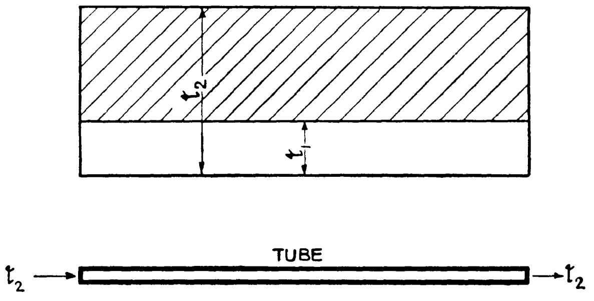

Fig. 109

The mean temperature difference shown by the shaded portion is clearly a constant and equal to $(t_2 - t_1)$.

Now let the feed water enter at a temperature t_f, and let the drain be cooled to temperature t_d. The diagram then is as shown in fig. 110.

The steam enters the tube at A, and during its travel from *a* to *b* the temperature remains sensibly constant. Actually there will be a slight drop, due to the presence of air in the steam and to the pressure drop along the tube. At *b* the whole of the steam will be condensed, and thereafter the temperature will drop from *b* to *c*. The feed water enters the evaporator at B and flows outside the tube, and the temperature rises from *d* to *e*; at *e* evaporation commences and continues at constant temperature t_1 to *f*.

It is generally best to separate out the surface necessary for feed heating from the surface necessary for evaporation, and provide a surface equal to the sum of the two. The mean temperature difference for the evaporating surface will be $(t_2 - t_1)$, and for the feed heating surface the difference may be found from Grashof's equation. The feed heating surface is small compared with the evaporating surface, and some designers allow for it by lowering the transmission rate slightly, and take the mean temperature difference as simply $(t_2 - t_1)$.

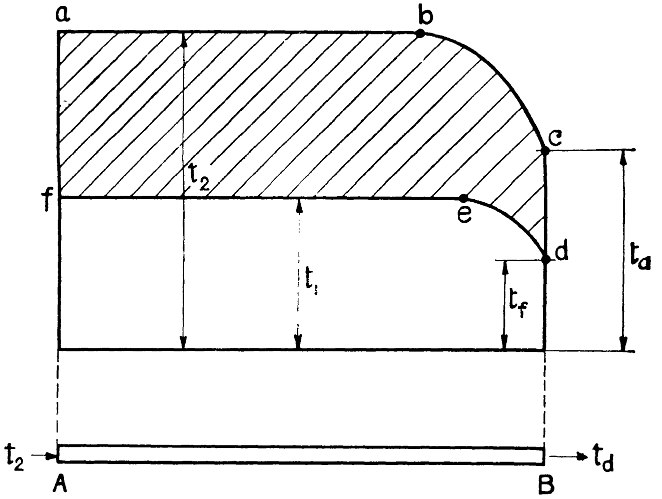

Fig. 110

Example 1.—An evaporator is to produce 50 tons of water per 24 hr. when supplied with raw feed water at a temperature of 60° F. The vapour produced is led to an auxiliary exhaust range at a pressure of 5 lb. per square inch gauge, and the steam supply is dry saturated steam at a pressure of 100 lb. per square inch gauge. Calculate the quantity of steam which must be supplied to the evaporator, and the surface necessary when copper coils are fitted.

$$\begin{aligned} &50 \text{ tons of water per 24 hr.} \\ &= 50 \times 2240/24 \\ &= 4666 \text{ lb. per hour.} \end{aligned}$$

Assuming $t_d = t_2$ in the equation (92), that is, the steam is discharged from the coils at steam temperature, the equation becomes

$$\begin{aligned} 0{\cdot}98Q_2 880 &= 4666(961{\cdot}3 + 227{\cdot}4 - 60), \\ \text{or} \quad Q_2 &= 4666 \times 1128{\cdot}7/(0{\cdot}98 \times 880) \\ &= 6106 \text{ lb. per hour.} \end{aligned}$$

This gives a ratio

$$\text{steam supplied/vapour produced} = 6106/4666$$
$$= 1\cdot3.$$

Separating out the evaporating and feed heating surfaces we have: For the evaporating surface the mean temperature difference θ_m is

$$\begin{aligned}\theta_m &= t_2 - t_1\\ &= 337\cdot9 - 227\cdot4\\ &= 110\cdot5^\circ\ \text{F}.\end{aligned}$$

Now from equation (94)

$$\begin{aligned}S_c &= H_F/(\theta_m R)\\ &= Q_1L_1/(\theta_m R).\end{aligned}$$

From fig. 108 $\quad R = 1180$ B.Th.U.

$$\begin{aligned}S_c &= 4666 \times 961\cdot3/(110\cdot5 \times 1180)\\ &= 34\cdot44\ \text{sq. ft.}\end{aligned}$$

For the feed heating surface we have

$$\begin{aligned}\theta_m &= \{(t_2 - t_f) - (t_2 - t_1)\}/[2\cdot3 \log\{(t_2 - t_f)/(t_2 - t_1)\}]\\ &= \{(337\cdot9-60)-(337\cdot9-227\cdot4)\}/[2\cdot3 \log\{(337\cdot9-60)/(337\cdot9-227\cdot4)\}]\\ &= 167\cdot4/(2\cdot3 \times 0\cdot4006)\\ &= 186^\circ\ \text{F}.\end{aligned}$$

Taking $R = 600$ B.Th.U. we get

$$\begin{aligned}S_H &= 4666(227\cdot4 - 60)/(186 \times 600)\\ &= 7\ \text{sq. ft.}\end{aligned}$$

The total surface therefore will be

$$\begin{aligned}S &= 34\cdot44 + 7\\ &= 41\cdot44\ \text{sq. ft.}\end{aligned}$$

Example 2.—Find the quantity of vapour which will be produced by an evaporator when supplied with 5000 lb. of exhaust steam per hour at a pressure of 4 lb. per square inch gauge, and having a dryness fraction of 0·9. The vapour is passed to a distiller wherein a vacuum of 15 in. Hg is maintained. The feed is raw water at a temperature of 100° F., and the drain from the evaporator coils has a temperature of 190° F. State also the ratio between the steam supplied and the vapour produced. (Neglect radiation losses.)

By equation (91), p. 209, we have

$$\begin{aligned}Q_2(qL_2 + t_2 - t_d) &= Q_1(L_1 + t_1 - t_f),\\ 5000(0\cdot9 \times 962\cdot4 + 224\cdot4 - 190) &= Q_1(990\cdot5 + 179 - 100),\\ \text{or}\qquad Q_1 &= 5000 \times 900\cdot56/1069\cdot5\\ &= 4210\ \text{lb. per hour.}\end{aligned}$$

The ratio will be

$$\text{steam supplied/vapour produced} = 5000/4210$$
$$= 1{\cdot}18.$$

Example 3.—During the test of an evaporator for a land power station the following figures were taken:

Shell pressure	5 lb. per square inch gauge.
Coil pressure	71·3 lb. per square inch gauge.
Temperature of coil drain	300° F.
Temperature of feed	60° F.
Temperature of steam entering coils	460° F.
Quantity of vapour produced ..	6025 lb. per hour.
Quantity of steam supplied ..	7450 lb. per hour.

Find the percentage of heat lost by radiation and state number of heat units lost per 24 hr.

In place of equation (93), p. 209, we write

$$xQ_2\{(H_s - H_2) + L_2 + t_2 - t_d\} = Q_1\{L_1 + t_1 - t_f\},$$

where x is the fraction of the heat used so that $(1 - x)$ will be lost.

By substituting the various numerical values in the above equation we get

$$\begin{aligned} x &= \{6025(961{\cdot}3+227{\cdot}4-60)\}/\{7450(1209-1183{\cdot}6+896{\cdot}4+317{\cdot}4-300)\} \\ &= 6025 \times 1128{\cdot}7/(7450 \times 939{\cdot}2) \\ &= 0{\cdot}97. \end{aligned}$$

The percentage of heat lost by radiation will therefore be

$$\begin{aligned} &= (1 - 0{\cdot}97)100 \\ &= 3 \text{ per cent.} \end{aligned}$$

The number of heat units lost per 24 hr. will be

$$\begin{aligned} &= 0{\cdot}03Q_2(H_s - H_2 + L_2 + t_2 - t_d)24 \\ &= 0{\cdot}03 \times 6250 \times 939{\cdot}2 \times 24 \\ &= 4{,}226{,}250 \text{ B.Th.U.} \end{aligned}$$

SEA-WATER EVAPORATORS

To prevent the density rising in the shell of an evaporator fed with sea water a brining apparatus is sometimes fitted, so that a portion of the brine is continuously withdrawn. The density of sea water is normally 1/32 and it is customary to limit the density in the shell to 3/32, this being the density at which precipitation commences. To keep the density down to this figure one-third of the feed water fed into the shell must be extracted as brine, while the remaining two-thirds are evaporated. The quantity of feed will therefore be one and a half times the production. The heat taken up in the evaporator will be the heat necessary to raise $1{\cdot}5Q_1$ lb. per hour from t_f to $t_1°$ F., and evaporate Q_1 lb. per hour into vapour at temperature t_1 and pressure P_1, or the heat taken up

$$= Q_1L_1 + \{1{\cdot}5Q_1(t_1 - t_f)\} \text{ B.Th.U. per hour.}$$

The heat exchange equation then becomes

$$0{\cdot}98Q_2(qL_2 + t_2 - t_d) = Q_1L_1 + \{1{\cdot}5Q_1(t_1 - t_f)\} \ldots\ldots (96)$$

The surface will be

$$S = 0{\cdot}98Q_2(qL_2 + t_2 - t_d)/(\theta_m R) = \{Q_1L_1 + 1{\cdot}5Q_1(t_1 - t_f)\}/(\theta_m R) \ldots (97)$$

Owing to the more rapid scaling which takes place in a sea-water evaporator, the coils require more frequent cleaning than they require in an evaporator fed with fresh water. As the scaling takes place the production falls off, consequent on the increased resistance to heat flow across the coils. The scale which adheres to the coils is a very bad conductor of heat and greatly retards the flow of heat. On board ship it is very important to maintain the evaporator production, and to achieve this the customary practice is to increase the steam pressure in the coils. This increased pressure gives an increase in the temperature difference θ_m, so that the production can be maintained till the end of a voyage. The scale found on the coils of a high-pressure sea-water evaporator is very much harder and more difficult to remove than that found on a low-pressure evaporator fed with the same sea water; this is due to the higher temperature of the high-pressure steam. The coils in the Weir evaporator shown in fig. 106 are arranged after the fashion of an Archimedian spiral, so that they tend to open out with heating and tend to coil up with cooling. This property is made use of with a view to breaking off the scale which adheres to the tubes. The procedure is to close the vapour valve and let the shell pressure rise till the safety valve blows; when this happens the coils will be at their maximum temperature, and consequently will be opened out to their fullest extent. The steam valve and all valves on the shell are thereafter closed. The blow-off cock is then opened and the water in the shell is blown into the sea. A partial vacuum will be formed in the shell due to the vapour condensing; cold sea water will then be drawn up through the blow-down pipe and into the shell, thereby increasing the vacuum, which results in a large volume of cold sea water entering the shell and surrounding the hot coils. This sudden cooling of the coils causes them to suddenly coil up, which results in much of the scale being broken off. This scale falls to the bottom of the evaporator and can be periodically removed through the cleaning door. An alternative method is to remove the front door, then turn on steam to the coils thereby heating them. The steam is thereafter shut off and a jet of cold water played on the tubes.

Example.—An evaporator for a merchant ship is to produce 60 tons of pure distilled water per 24 hr. from sea water at 50° F. The pressure of the steam in the coils is to be 80 lb. per square inch gauge, when the coils are clean, and the temperature of the coil drain is 5° F. lower than the entering steam temperature. The shell pressure is 5 lb. per square inch gauge. Calculate the surface necessary to give the above production, and calculate what the steam pressure will be when the coils are so scaled up that the transmission rate is reduced to three quarters its value with clean coils.

We shall treat this question by taking a slightly lower value of the transmission rate R than is given in fig. 108 for a shell pressure of 5 lb. per square inch gauge. From equation (97):

$$\begin{aligned} S &= \{Q_1L_1 + 1{\cdot}5Q_1(t_1 - t_f)\}/(\theta_m R); \\ Q_1 &= 60 \times 2240/24 = 5600 \text{ lb. per hour;} \\ L_1 &= 961{\cdot}3 \text{ from the steam tables;} \\ t_1 &= 227{\cdot}4^\circ \text{ F. from the steam tables;} \\ t_f &= 50^\circ \text{ F.;} \\ \theta_m &= (t_2 - t_1) = (323{\cdot}5 - 227{\cdot}4) = 96{\cdot}1^\circ \text{ F.} \end{aligned}$$

From fig. 108 $R = 1180$. Making this 5 per cent less we get

$$R = 1180 \times 0{\cdot}95 = 1121 \text{ B.Th.U.}$$

Substituting these values in the above equation we get

$$\begin{aligned} S &= (5600 \times 961{\cdot}3 + 1{\cdot}5 \times 5600 \times 177{\cdot}4)/(96{\cdot}1 \times 1121) \\ &= 6{,}375{,}600/(96{\cdot}1 \times 1121) \\ &= 59{\cdot}18 \text{ sq. ft.} \end{aligned}$$

When the transmission rate has fallen to three-quarters its original value R will be $1121 \times \frac{3}{4} = 840{\cdot}75$ B.Th.U.

$$\begin{aligned} \text{Then} \quad 59{\cdot}18 &= (5600 \times 961{\cdot}3 + 1{\cdot}5 \times 5600 \times 177{\cdot}4)/(\theta_m \times 840{\cdot}75), \\ \text{or} \quad \theta_m &= 6{,}375{,}600/(59{\cdot}18 \times 840{\cdot}75) \\ &= 128{\cdot}1^\circ \text{ F.} \end{aligned}$$

$$\begin{aligned} \text{Now} \quad \theta_m &= (t_2 - t_1), \\ \text{or} \quad t_2 &= 128{\cdot}1 + 227{\cdot}4 \\ &= 355{\cdot}5^\circ \text{ F.} \end{aligned}$$

From the steam tables the pressure corresponding to this temperature is approximately 144 lb. per square inch absolute.

CONTINUOUS BRINING OF SEA-WATER EVAPORATORS

It was stated on p. 214 that to maintain the density in an evaporator under 3/32 a brining apparatus was sometimes fitted. The method is known as continuous brining and is standard practice in naval vessels. Owing to the more general use of water-tube boilers in merchant ships, continuous brining is now rapidly becoming standard practice with this class of vessel also, and practically does away with the necessity for the regular cleaning of the coils.

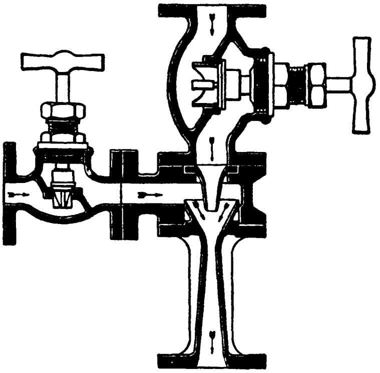
Fig. 111.—Weir Brine Ejector

The Admiralty method is to draw the brine from the evaporator by a ram pump, and discharge the brine overboard. To the suction pipe of this pump is fitted a brine cooler, which is supplied with a quantity of cold water from the discharge of the evaporator feed pump. The brine pump therefore deals with the brine in a diluted and cooled state, and by this means salting up of the valves and pump generally is prevented. The feed pump and the brine pump are driven by the same steam cylinder and therefore run at the same speed; the volumes of the two pumps are so proportioned that the relative quantities of feed and brine dealt with can be regulated to maintain the density at 3/32.

Merchant ship evaporators are commonly fitted with brine ejectors of the water-jet type. Fig. 111 shows the brine ejector as fitted to the Weir evaporator.

The operating water is cold sea water taken from the sanitary tank which is generally placed high up in the engine-room, so that sufficient head is obtained to work the ejector. The operating water is passed through a converging nozzle, and issues with a velocity commensurate with the height of the sanitary tank and the pressure between the brine valve and the nozzle exit. The high velocity jet draws the brine through the brine regulating valve and discharges it into the convergent-divergent diffuser nozzle, where a portion of the velocity energy is converted into pressure energy, thus enabling the ejector to discharge overboard.

The brine valve is relatively small and of conical shape, so that the velocity past the valve is high; this gives a scouring action to prevent the valve salting up, which would otherwise occur owing to the reduction in pressure and consequent self evaporation which takes place across this valve.

THEORY OF BRINE EJECTOR

The theory of this brine ejector is that of a jet pump dealing with a mixture of steam and water. Let fig. 112 represent the ejector.

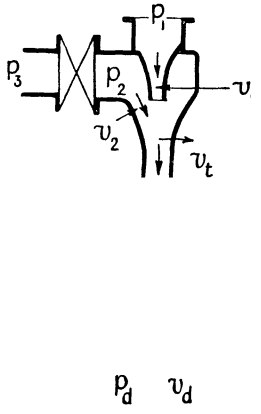

Fig. 112

Let p_1 be the operating water pressure in pounds per square inch absolute;
p_2, the pressure in the suction chamber per square inch absolute;
p_3, the evaporator shell pressure in pounds per square inch absolute;
p_d, the discharge pressure in pounds per square inch absolute;
t_1, the temperature of the operating water, degrees F.;
t_2, the temperature in the suction chamber, degrees F.;
t_3, the temperature of the brine, degrees F.; and
t_4, the temperature of the mixture being discharged, degrees F.

The cold water issuing from the nozzle condenses the steam formed by the reduction in pressure across the brine valve, that is, from p_3 to p_2, and cools the brine to the discharge temperature t_4. The steam is condensed on the surface of the cold operating jet. The temperature t_2 is dependent on the operating water temperature, and the surface of the jet presented to the steam in unit time.

If h_3 be the heat in the brine at pressure p_3, and H_2 total heat of the steam which has vaporized at pressure p_2, and h_2 the sensible heat of the brine at pressure p_2, we get, since the total heat per pound of brine before the brine value is the same as the total heat per pound of mixture of brine and steam after the brine value,

$$h_3 = xH_2 + (1 - x)h_2,$$

where x is the fraction of brine vaporized by the throttling at the brine valve.

If L_2 be the latent heat of the steam at pressure h_2 the above expression becomes

$$(t_3 - 32) = x(L_2 + t_2 - 32) + (1 - x)(t_2 - 32) \ldots\ldots\ldots (98)$$

Let v_1 be the velocity of efflux from the operating water nozzle in feet per second;
v_2, the velocity of the brine entering the diffuser nozzle in feet per second;
v_t, the velocity of the combined mass of brine and operating water at the throat section of the diffuser in feet per second; and
v_d, the velocity at the diffuser outlet in feet per second.

The effective pressure causing flow in the operating water nozzle is $(p_1 - p_2)$.

$$\therefore v_1 = C\sqrt{2g(p_1 - p_2)2{\cdot}3}, \ldots\ldots\ldots\ldots\ldots\ldots (99)$$

where C is the coefficient of velocity which varies between 0·9 and 0·95.

The velocity v_t may be found from the momentum equation, and if M_w is the mass of operating water in pounds per second and M_b the mass of brine extracted from the evaporator in pounds per second, the equation is

$$M_b v_2 + M_w v_1 = v_t(M_b + M_w),$$

$$\text{or} \quad v_t = (M_b v_2 + M_w v_1)/(M_b + M_w).$$

To allow for eddy losses which obtain in a practical ejector the above expression should be multiplied by a factor K, which varies between 0·8 and 0·9.

Then $$v_t = K\{(M_b v_2 + M_w v_1)/(M_b + M_w)\}. \quad\ldots\ldots\ldots(100)$$

Between the throat section and the discharge outlet the water velocity decreases and the pressure rises, so that by Bernoulli's theorem we have (neglecting the difference in level between throat and outlet)

$$(v_t^2/2g) + 2{\cdot}3p_2 = (v_d^2/2g) + 2{\cdot}3p_d.$$

The efficiency of the diverging portion of the diffuser nozzle is generally about 70 per cent. Inserting this value in the above expression we get

$$0{\cdot}7\{(v_t^2/2g) + 2{\cdot}3p_2\} = (v_d^2/2g) + 2{\cdot}3p_d. \quad\ldots\ldots\ldots(101)$$

If A_1 be the area in square inches of the operating water nozzle;
A_t, the area in square inches of the diffuser throat; and
A_d, the area in square inches of the diffuser outlet;

we have $$(M_w + M_b)/M_w = v_t A_t/(v_1 A_1),$$
or $$A_t/A_1 = v_1(M_w + M_b)/(v_t M_w),$$
and $$A_t/A_d = v_t/v_d.$$

Example 1.—A brine ejector is to withdraw a maximum of 1800 lb. of brine per hour from a marine evaporator and discharge the brine to sea. The shell pressure in the evaporator is 0·5 lb. per square inch gauge, and the pressure of the operating water is 20 lb. per square inch gauge. The sea level is 10 ft. above the ejector, and the water level in the evaporator is 2 ft. above the ejector. Calculate the quantity of operating water necessary, when the discharge velocity v_d is 5 ft. per second.

From equation (99) and assuming the pressure head of 0·5 lb. per square inch + 2 ft. of water head carried right up to the operating water nozzle exit,

$$\begin{aligned} v_1 &= 0{\cdot}9\sqrt{2g(p_1 - p_2)2{\cdot}3} \\ &= 0{\cdot}9\sqrt{64{\cdot}4\{(20 - 0{\cdot}5)2{\cdot}3 - 2\}} \\ &= 47{\cdot}27 \text{ ft. per second.} \end{aligned}$$

Since the maximum quantity of brine is being dealt with the brine valve will be full open, and the pressure due to the head of brine in the evaporator will be carried right up to the operating water nozzle exit. The velocity v_2 will therefore be very low, and, since we have in the equation for v_1 taken the pressure head right up to the operating water nozzle exit, the velocity v_2 may be taken as zero.

Then from equation (100) we have

$$v_t = K\{(M_b v_2 + M_w v_1)/(M_b + M_w)\},$$

i.e $$\begin{aligned} v_t &= K\{M_w v_1/(M_b + M_w)\} \\ &= 0{\cdot}85[(M_w \times 47{\cdot}27)/\{(1800/3600) + M_w\}] \\ &= 40{\cdot}17M_w/(0{\cdot}5 + M_w). \end{aligned}$$

From equation (101) we have

$$0{\cdot}7\{(v_t^2/2g) + 2{\cdot}3p_2\} = (v_d^2/2g) + 2{\cdot}3p_d,$$

or, neglecting the friction loss in the discharge pipe,

$$0{\cdot}7\{(v_t^2/2g) + 2{\cdot}3 \times 0{\cdot}5\} = (25/2g) + 10,$$
$$\text{or} \quad v_t = 30{\cdot}9 \text{ ft. per second.}$$

Substituting this value of v_t in the equation for M_w, we get

$$30{\cdot}9 = 40{\cdot}17M_w/(0{\cdot}5 + M_w),$$
$$\text{i.e.} \quad M_w = 1{\cdot}67 \text{ lb. per second}$$
$$= 6012 \text{ lb. per hour.}$$

Example 2.—During the test of a brine ejector the following figures were taken. Calculate the efficiency of the diffuser nozzle. The discharge velocity v_d was 4 ft. per second.

Pressure at inlet of operating water, 15 lb. per square inch.
Pressure in evaporator shell, 5 lb. per square inch.
Vacuum in ejector suction chamber, 6 in. Hg.
Pressure at discharge, 2 lb. per square inch.
Quantity of mixture of brine and operating water discharge, 9600 lb. per hour.
Quantity of operating water supplied, 7600 lb. per hour.

The water velocity v_1 will be, from equation (99),

$$v_1 = 0{\cdot}9\sqrt{2g(p_1 - p_2)2{\cdot}3}$$
$$= 0{\cdot}9\sqrt{64{\cdot}4(15 + 2{\cdot}94)2{\cdot}3}$$
$$= 46{\cdot}35 \text{ ft. per second.}$$

From equation (101) we have

$$E_d\{(v_t^2/2g) + 2{\cdot}3p_2\} = (v_d^2/2g) + 2{\cdot}3p_d,$$

where E_d is the diffuser efficiency;

$$\therefore E_d\{(v_t^2/2g) - 2{\cdot}3 \times 2{\cdot}94\} = (16/2g) + 2{\cdot}3 \times 2$$
$$E_dv_t^2 - 435E_d = 312.$$

The quantity of brine is

$$M_b = (9600 - 7600)/3600$$
$$= 0{\cdot}556 \text{ lb. per second.}$$

The quantity of operating water

$$M_w = 7600/3600$$
$$= 2{\cdot}11 \text{ lb. per second.}$$

From equation (100) we have

$$v_t = K\{(M_b v_2 + M_w v_1)/(M_b + M_w)\}.$$

Taking $v_2 = 0$, and $K = 0{\cdot}85$, this reduces to

$$\begin{aligned} v_t &= 0{\cdot}85\{(2{\cdot}11 \times 46{\cdot}35)/(0{\cdot}556 + 2{\cdot}11)\} \\ &= 31{\cdot}18 \text{ ft. per second.} \end{aligned}$$

Then substituting this value of v_t in

$$\begin{aligned} E_d v_t^2 - 435E_d &= 312 \\ E_d &= 312/537 \\ &= 0{\cdot}58, \end{aligned}$$

or the efficiency of the diffuser = 58 per cent.

CONTINUOUS FEED TO EVAPORATORS

It is very essential that the feed water fed to an evaporator should be continuous and controlled, and for this purpose all high-class evaporators are fitted with some form of water-level regulator. These regulators usually work on the float principle and maintain a constant water level inside the evaporator.

Fig. 113 shows the regulator as fitted to the Weir evaporator, and works on the float principle.

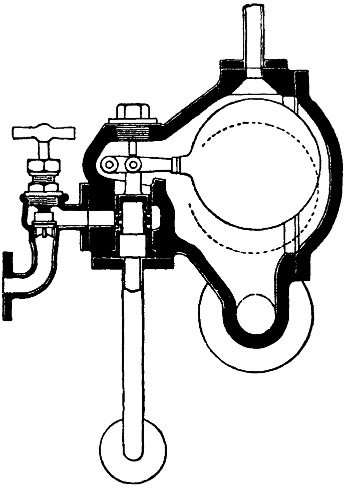

Fig. 113.—Weir Evaporator Water-level Regulator

The regulator is bracketed to the shell of the evaporator, and has a top connection to the steam dome of the evaporator, and a bottom connection to below the water level in the evaporator shell. This arrangement gives a still water level in the regulator, so that the float is not disturbed by the ebullition inside the shell, and a smooth working of the regulator is thereby ensured. The regulating or throttle valve consists of a hollow brass plug, working inside a brass sleeve; ports are cut in both plug and sleeve, so that as the float rises or falls these ports are put out of, or brought into register, thus controlling the quantity of water fed into the evaporator as required to maintain the water level constant. The feed water entering first passes the regulator stop valve, then through the ports in the sleeve and the regulating valve to the inside of the valve. It then flows downwards and enters the evaporator low down on the shell. The feed supply to the evaporator is generally taken from the sanitary tank which is continuously kept full by the sanitary pump, and the brine ejector and feed regulator are fed from the same pipe.

SAFETY VALVES ON EVAPORATORS

To all intents and purposes an H.P. evaporator is a steam boiler and is reckoned as such by the Board of Trade. This body has issued rules relative to the size of the safety valve to be fitted on the shell, so that in event of a tube bursting the pressure will not rise to a dangerous limit (see p. 225).

Evaporators have generally cast-iron shells which will not withstand boiler pressure. Boiler pressure would obtain if no safety valve were fitted and the vapour valve were inadvertently closed or a coil tube burst. To guard against such possibilities with H.P. evaporators, a restriction nozzle is fitted to the steam supply pipe or valve, which is of such a bore that the quantity of steam passing this nozzle would be passed by the safety valve when the shell pressure is not so high as to be dangerous.

Let a be the nozzle area in square inches;

- A, the safety valve area when the valve is open to its fullest extent (this is generally termed when the valve has popped);
- P, the maximum steam supply pressure in pounds per square inch absolute; and
- p, the safety shell pressure in pounds per square inch absolute.

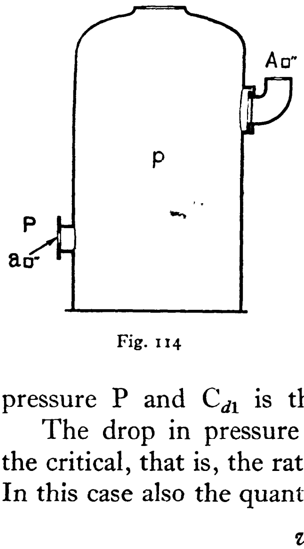

Fig. 114

It is assumed that in the event of a tube bursting the burst is of such a nature that it causes no restriction to the flow of steam. The assumed conditions therefore will be as shown in fig. 114, that is, the steam passes the nozzle without restriction directly into the shell.

If the safety pressure p is below the critical pressure for P (as is generally the case), then the quantity of steam passing the nozzle will be

$$w = C_{d1} 0{\cdot}3155 a \sqrt{P/V_1} \text{ lb. per second,..(102)}$$

where V_1 is the specific volume of the steam at pressure P and C_{d1} is the coefficient of discharge for the nozzle.

The drop in pressure across the safety valve is generally slightly over the critical, that is, the ratio of atmospheric pressure to p is less than 0·5457. In this case also the quantity of steam which will pass the safety valve is

$$w = C_{d2} 0{\cdot}3155 A \sqrt{p/V_2},$$

where V_2 is the specific volume of the steam at pressure p, and C_{d2} is the coefficient of discharge for the safety valve.

These two quantities of steam must be equal for settled conditions, so that

$$C_{d1} 0{\cdot}3155 a \sqrt{P/V_1} = C_{d2} 0{\cdot}3155 A \sqrt{p/V_2},$$

$$\text{or } \quad a/A = (C_{d2}/C_{d1}) \sqrt{pV_1/PV_2} \ldots\ldots\ldots(103)$$

The area (a) of the nozzle must be such that the steam quantity necessary for normal working of the evaporator will pass, and for this purpose the evaporator should be designed to produce its rated output at about 100 lb. per square inch. The pressure drop across the restriction nozzle under normal working conditions will then be from the boiler pressure to 100 lb. per square inch. If P_w be the working pressure in the coils, and W the quantity of steam in pounds per second necessary for normal working of the evaporator, then as P_w is below the critical pressure the minimum area of the nozzle must be that given by the following expression.

$$W = C_{d1} 0{\cdot}3155 a \sqrt{P_w/V_1}, \ldots\ldots\ldots\ldots\ldots (104)$$

$$\text{or} \qquad a = (W/C_{d1} 0{\cdot}3155)\sqrt{V_1/P_w}. \ldots\ldots\ldots\ldots (105)$$

Example 1.—The restriction nozzle as fitted to a marine evaporator is to pass 5000 lb. of steam per hour for normal working. The steam supply pressure is 250 lb. per square inch gauge, and the coil pressure is 100 lb. per square inch gauge. Calculate the area of the nozzle required, assuming a coefficient of discharge of 0·65 for a sharp edged nozzle, and the steam supplied as being dry and saturated.

From equation (105) we have

$$\begin{aligned} a &= (W/C_{d1} 0{\cdot}3155)\sqrt{V_1/P_w}; \\ \text{and} \quad W &= 5000/3600 = 1{\cdot}389 \text{ lb. per second,} \\ P &= 264{\cdot}7 \text{ lb. per square inch absolute,} \\ V_1 &= 1{\cdot}75 \text{ c. ft. per pound (from the steam tables).} \\ \therefore a &= \{1{\cdot}389/(0{\cdot}65 \times 0{\cdot}3155)\}\sqrt{1{\cdot}75/264{\cdot}7} \\ &= 0{\cdot}54 \text{ sq. in.} \end{aligned}$$

CONDITIONS WHEN A TUBE BURSTS

When a tube bursts in an evaporator it is assumed that the flow from the restriction nozzle into the evaporator shell is unimpeded. Assuming this condition we have that the steam flowing through the restriction nozzle has its velocity increased, and the total heat in the steam as it leaves the restriction nozzle will be lowered by an amount equal to the increase in the velocity energy possessed by the steam as it leaves the nozzle. The steam thereafter will pass to the evaporator shell unimpeded, where the velocity or kinetic energy will all be frittered down into heat. On the whole no work will be done, so that the total heat of the steam in the evaporator shell will be equal to that possessed by the steam before it entered the restriction nozzle. This neglects the small amount of kinetic energy in the steam before it entered the nozzle. The condition of the steam in the evaporator shell is generally best found from the total heat entropy chart, by reading horizontally from the total heat in the steam supplied to the shell pressure curve, when the condition may read off.

Example 2.—Taking the evaporator given in example 1, find the steam temperature in the shell when a tube has burst and the safety valve is blowing off at a pressure of 15 lb. per square inch gauge.

The total heat of dry saturated steam at 250 lb. per square inch gauge pressure is, from the steam tables, 1202·3; moving horizontally, on the HΦ chart, from this total heat till we arrive at the 30 lb. per square inch absolute curve, we read the superheat to be 80° F., so that the steam temperature will be $250 + 80 = 330°$ F. The steam in the evaporator shell will therefore be at a pressure of 30 lb. per square inch absolute and at a total temperature of 330° F.; these are the steam supply conditions to the safety valve, and from the steam tables the volume V_2 of this steam may be found.

Example 3.—Let it be required to find the area through the safety valve for the foregoing evaporator.

From equation (103) we have

$$a/A = (C_{d2}/C_{d1})\sqrt{pV_1/PV_2}.$$

Now p is 30 lb. per square inch absolute;
P, 265 lb. per square inch absolute;
C_{d2}, say 0·6 for the safety valves;
C_{d1}, 0·65 as before;
V_1, 1·75 c. ft. as before;
V_2, 15·5 c. ft. (from the steam tables for 30 lb. per square inch and 330° F.); and
a, 0·54 sq. in.

Then, substituting these values in the above equation, we get

$$\begin{aligned} 0{\cdot}54/A &= (0{\cdot}6/0{\cdot}65)\sqrt{(30 \times 1{\cdot}75)/(265 \times 15{\cdot}5)}, \\ \text{or} \qquad A &= 0{\cdot}54 \times 0{\cdot}65/(0{\cdot}6 \times 0{\cdot}1132) \\ &= 5{\cdot}16 \text{ sq. in.} \end{aligned}$$

If the diameter of the safety valve be $5\frac{1}{2}$ in. then the lift would be 0·262 in.

When the pressure drop across the safety valve is less than the critical, that is, if the ratio of atmospheric pressure to the safety shell pressure is less than 0·5457, the steam flow past the safety valve is best found from the heat drop.

Let H be the total heat in the steam inside the shell, and for the assumed conditions of free flow across the burst tube, will equal the total heat in the steam in front of the restriction nozzle;
h, the total heat in the steam as it is discharged past the safety faces;
J, Joule's equivalent = 778 B.Th.U.; and
v, the maximum steam velocity past the safety valve faces;

$$\begin{aligned} \text{Then} \qquad v^2/(2g) &= J(H - h), \\ \text{or} \qquad v &= \sqrt{2gJ(H-h)}, \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (106) \\ \text{and} \qquad w &= (C_{d2}A/144V_3)\sqrt{2gJ(H-h)}. \qquad \ldots\ldots\ldots\ldots \quad (107) \end{aligned}$$

The value of $H - h$ may be read off directly from the HΦ chart, and V_3, the volume per pound of the steam as it leaves the safety valve faces, may be deduced from the conditions of the steam at that point as shown on the HΦ chart.

Example 4.—The total heat in the steam in the evaporator shell is 1247 B.Th.U., and the shell pressure is 20 lb. per square inch absolute. Calculate the necessary area past the safety valve faces to pass 6000 lb. per hour. Take the coefficient of discharge as 0·56.

From equation (107) we have

$$\begin{aligned} w &= \{C_{d2}A/(144V_3)\}\sqrt{2gJ(H-h)}, \\ &= 6000/3600 \\ &= 1{\cdot}668 \text{ lb. per second,} \\ C_{d2} &= 0{\cdot}56. \end{aligned}$$

The ratio 14·7/20 = 0·735, so that the heat drop is within the critical range and will therefore be from 20 lb. to 14·7 lb. per square inch. To find the heat drop $(H-h)$, drop a vertical on the HΦ chart from the intersection of the 1247 horizontal, and the 20 lb. pressure curve on to the 14·7 lb. pressure curve, and read the total heat h on the lower horizontal. The value of $(H-h)$ thus found is $(1247-1218) = 29$ B.Th.U.

The value of V_3 is found from the condition of the steam at the lower horizontal on the HΦ chart. At the intersection of the lower horizontal and the 14·7 lb. pressure curve read along the temperature curve, and we read the superheat as 145° F. The total steam temperature will therefore be the saturation temperature at 14·7 lb. per square inch plus 145, or $212 + 145 = 357°$ F. From the steam tables we find that steam at 14·7 lb. per square inch absolute and 357° F. has a volume of 33 c. ft.; i.e. $V_3 = 33$.

Then, substituting these values in the above, we get

$$\begin{aligned} 1{\cdot}668 &= \{0{\cdot}56A/(33\times144)\}\sqrt{64{\cdot}4\times778\times29}, \\ \text{or}\quad A &= 1{\cdot}668\times33\times144/(0{\cdot}56\times1205) \\ &= 11{\cdot}74 \text{ sq. in.} \end{aligned}$$

BOARD OF TRADE RULE FOR THE SIZE OF EVAPORATOR SAFETY VALVES

The rule issued by the Board of Trade for the relation between the area of the restriction nozzle and the area of the safety valve for marine evaporators is as follows:

$$a = Ap/(6P),$$

where a is the area of the restriction nozzle in square inches (the nozzle is sharp edged);

A, the combined area of the safety valves;

p, the working shell pressure in the evaporator in pounds per square inch absolute; and

P, the pressure of the steam supply to the evaporator in pounds per square inch absolute.

TESTING BRINE DENSITY

The scale of brine density used for evaporators is in thirty-seconds. This is merely a convention, and originated in consequence of the fact that ordinary sea water at a temperature of 200° F. contains about 1/32 part of its weight of matter in solution. The density of sea water therefore at a temperature of 200° F. is 1/32; this density increases as the temperature is reduced, and for comparative purposes the density is usually referred to 200° F. as a standard.

On this same scale ordinary towns' fresh water has a density 0.

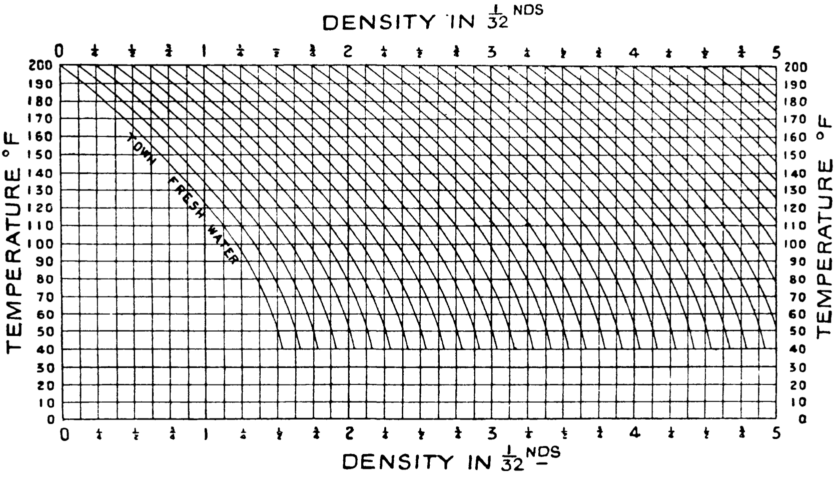

Fig. 115.—Weir Density-Temperature Chart for Brine

To determine the brine density in an evaporator a small portion of the brine is first withdrawn into an open pot. A hydrometer graduated in thirty-seconds is then floated in the brine and the density noted; the temperature must also be noted by placing a thermometer in the brine. The density thus found must then be corrected to a temperature of 200° F., and to perform the correction the curves shown in fig. 115 may be used. As an example, let the density and temperature of the brine as withdrawn from an evaporator be $3\frac{1}{4}/32$ and 110° F. respectively. From the intersection of the 110° F. horizontal and the $3\frac{1}{4}$ vertical on the chart travel up the curve to the top horizontal, and read the density corrected to 200° F. as $2\frac{1}{8}/32$.

MULTIPLE EFFECT EVAPORATORS

In the multiple effect evaporator we have two or more evaporators in series, known as the first, second, third, &c., effects. The vapour from the first passes to the heating coils of the second and thus becomes the heating steam for the second, and the vapour from the second passes to the heating coils of the third, and so on for any number of effects. The same process is adopted for the coil drains, i.e. the coil drain from the first effect passes to the heating coils of the second effect, and the coil drain from the second effect passes to the heating coils of the third effect, and so on. The same process is also adopted for the entering feed water. All the feed water enters the first effect, a portion is evaporated and the remainder passes to the second effect where a portion is evaporated and the remainder passes to the third effect, and so on. Feed heaters are generally placed in the vapour pipes between the effects, and the feed water is passed through these and progressively heated from the last to the first effects. The arrangement is an efficient one, the efficiency increasing with the number of effects; the first cost of the installation, however, also increases with the number of effects. For power station work the number of effects seldom exceeds three. The comparative ratios of the amount of fresh water produced per pound of steam used for single, double, and triple effects may be taken as follows:

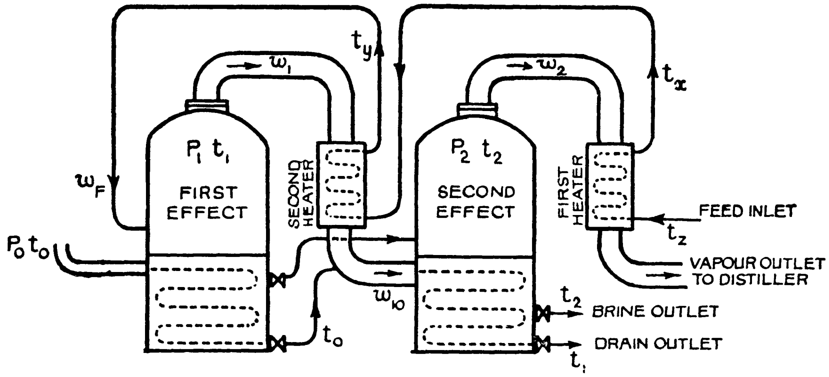

Fig. 116.—Double-effect Evaporating Plant

Single Effect	Double Effect	Triple Effect
0·8	1·72	2.

Let fig. 116 represent in diagrammatic form a double-effect evaporating plant.

Let P_0 be the entering steam pressure in pounds per square inch;

t_0, the entering steam temperature in pounds per square inch (saturated);

P_1, the shell pressure in the first effect;

t_1, the shell temperature in the first effect;

P_2, the shell pressure in the second effect;

t_2, the shell temperature in the second effect;

t_z, the inlet temperature of the feed water;

t_x, the outlet temperature of the feed water from first heater;

t_y, the outlet temperature of the feed water from the second heater;

w_0, the weight of steam entering the first effect heating coils in pounds per hour;

w_1, the weight of vapour produced in the first effect in pounds per hour;
w_{10}, the weight of vapour entering the heating coils of the second effect in pounds per hour;
w_2, the weight of vapour produced in the second effect in pounds per hour;
W, the total weight of vapour produced, i.e. $(w_1 + w_2)$;
L_0, the latent heat of the steam at pressure P_0;
L_1, the latent heat of the vapour at pressure P_1;
L_2, the latent heat of the vapour at pressure P_2; and
w_F, the quantity of feed water in pounds per hour

We shall assume in what follows that the steam which is condensed in the coils is just condensed, that is, the drain from the coils is at steam temperature, and that there is no radiation loss. Then the quantity of heat which flows across the coil surface in the first effect is

$$w_0 \times L_0.$$

This quantity of heat is absorbed by the entering feed water in raising its temperature from t_y to t_1 and in evaporating the quantity w_1, and is

$$w_F(t_1 - t_y) + w_1 L_1.$$
$$\therefore\ w_0 L_0 = w_F(t_1 - t_y) + w_1 L_1.$$

This equation may be called the heat exchange equation in the first effect, and may be written thus

$$w_1 = \{w_0 L_0 - w_F(t_1 - t_y)\}/L_1 \ldots\ldots\ldots\ldots\ldots(108)$$

A similar equation may be worked out for the second effect.

The weight of vapour entering the heating coils of the second effect is w_1 less the vapour condensed in the second heater.

The heat exchange equation for the second heater is

$$(w_1 - w_{10})L_1 = w_F(t_y - t_x),$$
$$\text{or} \quad w_{10} = w_1 - \{w_F(t_y - t_x)\}/L_1 \ldots\ldots\ldots(109)$$

The heat transference in the second effect is composed of the heat given up by the condensation of w_{10} lb. of steam and the cooling of w_0 lb. of drain from the first effect from t_0 to t_1, and is therefore

$$w_{10} L_1 + w_0(t_0 - t_1) \ldots\ldots\ldots\ldots\ldots\ldots(110)$$

This transference of heat will give an evaporation of w_{20} lb. of vapour, so that the heat exchange equation for the second effect will be

$$w_{10} L_2 + w_0(t_0 - t_1) = w_{20} L_2, \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(111)$$
$$\text{or} \quad w_{20} = \{w_{10} L_2 + w_0(t_0 - t_1)\}/L_2 \ldots\ldots(112)$$

The feed water entering the second effect is at a temperature t_1 which is higher than t_2, so that a portion of the entering feed will flash into steam; this is sometimes spoken of as self evaporation. The quantity of steam thus evaporated is

$$w_{40} = \{(w_F - w_1)(t_1 - t_2)\}/L_2 \ldots\ldots\ldots\ldots (113)$$

The total vapour in the second effect is w_2,

$$\begin{aligned} \therefore\ w_2 &= w_{20} + w_{40}, \\ &= [\{w_{10}L_2 + w_0(t_0 - t_1)\}/L_2] + [\{(w_F - w_1)(t_1 - t_2)\}/L_2]. \end{aligned}$$

The production efficiency is

$$\begin{aligned} &= \text{weight of vapour produced/weight of steam used} \\ &= \{w_1 + w_2\}/w_0 \\ &= [\{w_0L_0 - w_F(t_1 - t_y)\}/(L_1w_0)] \\ &\qquad + [\{w_{10}L_2 + w_0(t_0 - t_1) + (w_F - w_1)(t_1 - t_2)\}/L_2w_0] \ldots (114) \end{aligned}$$

$$\begin{aligned} \text{Now}\quad w_{10} &= w_1 - [\{w_F(t_y - t_x)\}/L_1], \\ \text{and}\quad w_1 &= \{w_0L_0 - w_F(t_1 - t_y)\}/L_1, \\ \therefore\ w_{10} &= [\{w_0L_0 - w_F(t_1 - t_y)\}/L_1] - [\{w_F(t_y - t_x)\}/L_1]. \end{aligned}$$

Then, substituting these values for w_{10} and w_1 in equation (114), it becomes on reduction

$$\begin{aligned} &= [(2/L_1)\{w_0L_0 - w_F(t_1 - t_y)\} - w_F(t_y - t_x)] + [\{w_0(t_0 - t_1) + w_F(t_1 - t_2)\}/L_2] \\ &\qquad - [(t_1 - t_2)/L_2][\{w_0L_0 - w_F(t_1 - t_y)\}/(L_1L_2)] \ldots\ldots\ldots\ldots (115) \end{aligned}$$

Example.—A double effect evaporating plant has a first effect shell pressure of 5 lb. per square inch gauge, and a second effect shell pressure of 5 in. Hg vacuum. The steam supply pressure is 20 lb. per square inch gauge. The feed water entering the first effect is at a temperature of 190° F., and the temperature rise in the second heater is 90° F. The quantity of feed entering the first effect is equal to the steam entering the first effect. Calculate the production efficiency.

From equation (115) we have

Production efficiency

$$\begin{aligned} &= [(2/L_1)\{w_0L_0 - w_F(t_1 - t_y)\} - w_F(t_y - t_x)] + [\{w_0(t_0 - t_1) + w_F(t_1 - t_2)\}/L_2] \\ &\qquad - [(t_1 - t_2)/L_2][\{w_0L_0 - w_F(t_1 - t_y)\}/(L_1L_2)]. \end{aligned}$$

Now $w_0 = 1$.

L_0 is the latent heat of entering steam at 20 lb. per square inch gauge pressure, i.e. 940 B.Th.U.

$w_F = 1$. (Since the feed water equals the steam used.)

t_1 is the temperature corresponding to 5 lb. per square inch gauge, i.e. 228° F.

t_y is the temperature of entering feed water, i.e. 190° F.

t_x is the temperature of feed water entering the second heater, i.e. $(190 - 90) = 100°$ F.
t_0 is the steam temperature corresponding to pressure of entering steam, i.e. 259° F.
t_2 is the vapour temperature in second effect, or temperature corresponding to 5 in. Hg vacuum, i.e. 203° F.
L_2 is the latent heat of vapour in second effect at 5 in. Hg vacuum, i.e. 976 B.Th.U.
L_1 is the latent heat of vapour in first effect at 5 lb. per square inch gauge, or 960·5 B.Th.U.

Then, substituting these numerical values in the production efficiency equation, we get

Production efficiency

$$\begin{aligned} &= [(2/960{\cdot}5)\{940 - (228 - 190)\} - (190 - 100)] \\ &\qquad + [\{(259 - 228) + (228 - 203)\}/976] \\ &\qquad - [(228 - 203)/976][\{940 - (228 - 190)\}/(960{\cdot}5 \times 976)] \\ &= 1{\cdot}69 + 0{\cdot}0574 - 0{\cdot}00002 \\ &= 1{\cdot}75. \end{aligned}$$

HEATING SURFACES TO BE PROVIDED IN MULTIPLE EFFECT EVAPORATORS

The curve given in fig. 108 shows that the coefficient of heat transmission or the heat transmission rate varies with the shell pressure. The allowable rate, therefore, will be different for each effect. When a multiple effect evaporator is working the various shell pressures adjust themselves, so that the transmission rate in each effect is the maximum for the conditions. From considerations of manufacture it is generally the practice to make the heating surface of each effect the same.

Let R_1 be the transmission rate in the first effect;
R_2, the transmission rate in the second effect;
S_1, the heating surface in the first effect;
S_2, the heating surface in the second effect;
H_{R1}, the heat transference in first effect in B.Th.U. per hour; and
H_{R2}, the heat transference in second effect in B.Th.U. per hour.

Then
$$S_1 = H_{R1}/(R_1\theta_1), \tag{116}$$
and
$$S_2 = H_{R2}/(R_2\theta_2), \tag{117}$$

where θ_1 and θ_2 are the temperature differences in the first and second effects respectively.

If S_1 is to be equal to S_2,

then
$$H_{R1}/(R_1\theta_1) = H_{R2}/(R_2\theta_2),$$
or
$$H_{R1}/H_{R2} = R_1\theta_1/(R_2\theta_2). \tag{118}$$

Now $$H_{R1} = w_1L_1 + w_F(t_1 - t_y), \qquad (119)$$
and $$H_{R2} = w_2L_2 - (w_F - w_1)(t_1 - t_2) \qquad (120)$$

Then, substituting these values of H_{R1} and H_{R2} in equation (118), we get

$$\{w_1L_1 + w_F(t_1 - t_y)\}/\{w_2L_2 - (w_F - w_1)(t_1 - t_2)\} = R_1\theta_1/(R_2\theta_2) \qquad (121)$$

It is general practice and is sufficiently near for practical purposes to assume values for θ_1 and θ_2, by dividing the total temperature drop from the steam inlet temperature to the vapour outlet temperature from the last effect, and dividing this drop equally among the effects.

For a double effect then θ_1 will equal θ_2 or $\theta_1 = \theta_2 = (t_0 - t_2)/2$.

This fixes the shell temperatures and pressures, and R_1 and R_2 may be found from fig. 108 for the corresponding shell pressure. This leaves w_1 and w_2 unknown in equation (121). The total production of the plant is

$$W = (w_1 + w_2) \qquad (122)$$

w_1 and w_2 can therefore be found from these two equations, (121) and (122). When w_1 and w_2 are found the heat flows H_{R1} and H_{R2} can be found and the surface S_1 or S_2 found from equation (116) or (117).

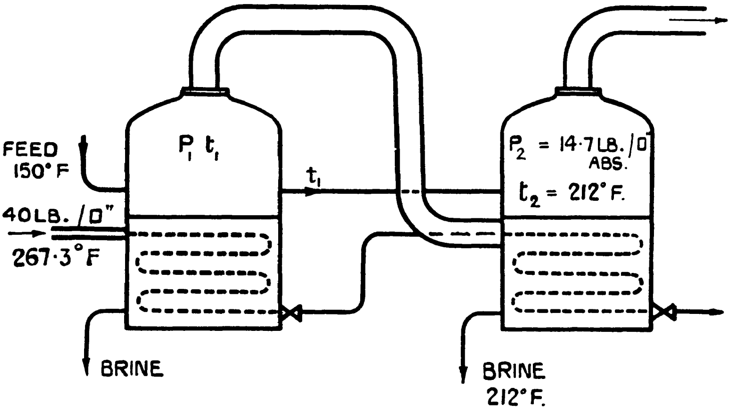

Fig. 117

Example.—A double-effect evaporating plant is to produce 100 tons per 24 hours of fresh water from sea water, which enters the first effect at 150° F. The available steam pressure is 40 lb. per square inch absolute. The heating coils are of copper, and there are no feed heaters. Assuming that the temperature drop is equally divided between the two effects, estimate what the probable shell pressures will be, and find the necessary surface in each effect.

Fig. 117 illustrates the plant.

The total temperature range or drop from the supply steam temperature to the temperature of the vapour leaving the second effect is

$$(267{\cdot}3 - 212) = 55{\cdot}3^\circ \text{ F.}$$

This temperature drop is divided equally between the two effects, so that

$$\begin{aligned} t_1 &= 267{\cdot}3 - 55{\cdot}3/2 \\ &= 267{\cdot}3 - 27{\cdot}65 \\ &= 239{\cdot}65^\circ \text{ F.} \end{aligned}$$

The pressure corresponding to this temperature is 25 lb. per square inch absolute.

The shell pressures in the first and second effects will therefore be 25 and 14·7 lb. per square inch respectively.

From fig. 108 we get for these shell pressures 25 and 14·7 lb. per square inch, $R_1 = 1250$ and $R_2 = 1030$ B.Th.U. respectively.

Then
$$\begin{aligned} R_1\theta_1 &= 1250 \times 27{\cdot}65 \\ &= 34{,}562, \end{aligned}$$
and
$$\begin{aligned} R_2\theta_2 &= 1030 \times 27{\cdot}65 \\ &= 28{,}479. \end{aligned}$$
Now
$$H_{R1} = w_1L_1 + \tfrac{3}{2}W(t_1 - t_y).$$

The feed water being sea water the quantity of feed entering the evaporator w_F will be $\tfrac{3}{2}W$ to keep the density under 3/32.

Also
$$\begin{aligned} H_{R2} &= w_2L_2 - (w_F - w_1)(t_1 - t_2) \\ &= w_2L_2 - (\tfrac{3}{2}W - w_1)(t_1 - t_2). \end{aligned}$$

Then from equation (121) we have

$$\{w_1L_1 + \tfrac{3}{2}W(t_1 - t_y)\}/\{w_2L_2 - (\tfrac{3}{2}W - w_1)(t_1 - t_2)\} = R_1\theta_1/(R_2\theta_2).$$

By substitution in this equation we get

$$\begin{aligned} &[w_1 \times 925 + \{3 \times 100 \times 2240/(24 \times 2)\}(239{\cdot}65 - 150)] \\ &\quad \div [w_2 \times 970{\cdot}4 - \{3 \times 100 \times 2240/(24 \times 2) - w_1\}(239{\cdot}65 - 212)] \\ &= (1250 \times 27{\cdot}65)/(1030 \times 27{\cdot}65); \end{aligned}$$

or
$$\begin{aligned} (925w_1 + 14{,}000 \times 89{\cdot}65)/\{970{\cdot}4w_2 - (14{,}000 - w_1)27{\cdot}65\} &= 1{\cdot}213, \\ 891{\cdot}56w_1 - 1177w_2 &= -\,1{,}724{,}652, \end{aligned}$$
and
$$\begin{aligned} w_1 + w_2 &= 100 \times 2240/24 \\ &= 9333. \end{aligned}$$
Therefore
$$\begin{aligned} 891{\cdot}56(9333 - w_2) - 1177w_2 &= -\,1{,}724{,}652, \\ \text{i.e.} \qquad -\,2068{\cdot}56w_2 &= -\,10{,}045{,}221, \\ \therefore \quad w_2 &= 5000 \text{ lb. per hour.} \end{aligned}$$
And
$$\begin{aligned} w_1 &= 9333 - 5000 \\ &= 4333 \text{ lb. per hour.} \end{aligned}$$
Now
$$\begin{aligned} H_{R1} &= w_1L_1 + \tfrac{3}{2}W(t_1 - t_y) \\ &= 4333 \times 891{\cdot}56 + \tfrac{3}{2} \times 9333(239{\cdot}65 - 150) \\ &= 5{,}120{,}184 \text{ B.Th.U. per hour.} \end{aligned}$$

$$\begin{aligned}
\text{And}\qquad S_1 &= H_{R1}/(R_1\theta_1)\\
&= 5{,}120{,}184/34{,}562\\
&= 148 \text{ sq. ft.}
\end{aligned}$$

$$\begin{aligned}
\text{Again}\qquad H_{R2} &= w_2L_2 - (\tfrac{3}{2}W - w_1)(t_1 - t_2)\\
&= 5000 \times 970{\cdot}4 - (\tfrac{3}{2} \times 9333 - 4333)(239{\cdot}65 - 212)\\
&= 4{,}792{,}111 \text{ B.Th.U. per hour.}
\end{aligned}$$

$$\begin{aligned}
\text{And}\qquad S_2 &= H_{R2}/(R_2\theta_2)\\
&= 4{,}792{,}111/28{,}479\\
&= 168 \text{ sq. ft.}
\end{aligned}$$

Example.—A sea-water, double effect evaporating plant, fig. 118, is supplied with saturated steam at 20 lb. per square inch gauge. The first effect shell is under a vacuum of 10 in. Hg, and the second effect is under a vacuum of 26 in. Hg. The feed water enters the first feed heater at 50° F. and has its temperature increased 35° F. in each heater. The heating surface is 250 sq. ft. in each effect made up of copper tube coils, having a thickness of 10 S.W.G. Calculate the output from each effect, and find the production efficiency.

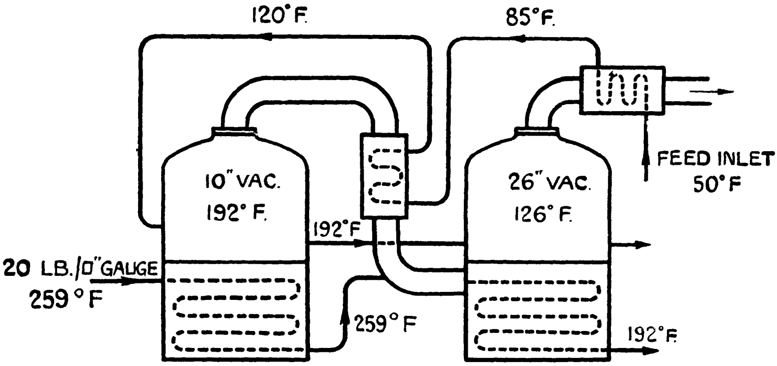

Fig. 118

The feed water entering the evaporator will be

$$\begin{aligned}
w_F &= \tfrac{3}{2}W,\\
\text{or}\qquad w_F &= \tfrac{3}{2}(w_1 + w_2),\\
\theta_1 &= 259 - 192\\
&= 67^\circ \text{ F.};
\end{aligned}$$

and from fig. 108

$$R_1 = 790 \text{ B.Th.U.}$$

Now

$$\begin{aligned}
\theta_2 &= 192 - 126\\
&= 66^\circ \text{ F.};
\end{aligned}$$

and from fig. 108

$$R_2 = 350 \text{ B.Th.U.}$$

The total transference of heat in the first effect H_{R1} is, from equation (119),

$$H_{R1} = w_1L_1 + \tfrac{3}{2}W(t_1 - t_y);$$

and from equation (116)

$$\begin{aligned} S_1 &= H_{R1}/(R_1\theta_1); \\ \therefore\ w_1L_1 + \tfrac{3}{2}W(t_1 - t_y) &= S_1R_1\theta_1, \\ \text{or}\quad 982w_1 + \tfrac{3}{2}W(192 - 120) &= 250 \times 790 \times 67, \\ \text{i.e.}\quad 982w_1 + 108W &= 13{,}230{,}000. \\ \text{Now}\quad W &= w_1 + w_2; \\ \therefore\ 982w_1 + 108w_1 + 108w_2 &= 13{,}230{,}000, \\ \text{or}\quad 1090w_1 + 108w_2 &= 13{,}230{,}000 \ldots\ldots\ldots\ldots\ldots(123) \end{aligned}$$

Again the total transference of heat in the second effect is found from equation (120):

$$\begin{aligned} H_{R2} &= w_2L_2 - (w_F - w_1)(t_1 - t_2), \\ \text{or}\quad w_2L_2 - (w_F - w_1)(t_1 - t_2) &= S_2R_2\theta_2, \\ 1021w_2 - (\tfrac{3}{2}W - w_1)(192 - 126) &= 250 \times 350 \times 66, \\ 1021w_2 - 99W + 66w_1 &= 5{,}77{,}000, \\ \text{or, since}\ W &= w_1 + w_2, \\ 1021w_2 - 99w_1 - 99w_2 + 66w_1 &= 5{,}770{,}000, \\ 922w_2 - 33w_1 &= 5{,}770{,}000. \ldots\ldots\ldots\ldots\ldots(124) \end{aligned}$$

Then equations (123) and (124) are

$$\left.\begin{aligned} 1090w_1 + 108w_2 &= 13{,}230{,}000 \\ -\,33w_1 + 922w_2 &= 5{,}770{,}000 \end{aligned}\right\}$$

Solving these simultaneous equations we get

$$\begin{aligned} w_1 + 0{\cdot}992w_2 &= 12{,}137 \\ -\,w_1 + 27{\cdot}9\ w_2 &= 175{,}000 \\ \hline 28{\cdot}892w_2 &= 187{,}137 \\ w_2 &= 6477 \text{ lb. per hour,} \\ \text{and}\quad w_1 &= 12{,}137 - 0{\cdot}992 \times 6477 \\ &= 5712 \text{ lb. per hour.} \end{aligned}$$

The total production or output will then be

$$\begin{aligned} W &= 6477 + 5712 \\ &= 12{,}189 \text{ lb. per hour,} \\ \text{or}\quad &= 12{,}189 \times 24/2240 \\ &= 130{\cdot}6 \text{ tons per 24 hr.} \end{aligned}$$

The steam used in the coils of the first effect is w_0, and the heat given up is w_0L_0. This equals the heat transference,

$$\text{or} \quad w_0L_0 = w_1L_1 + \tfrac{3}{2}W(t_1 - t_y),$$

$$\text{or} \quad w_0 = \{5712 \times 982 + \tfrac{3}{2} \times 12{,}137\,(192 - 120)\}/940$$

$$= (5{,}610{,}000 + 1{,}310{,}800)/940$$

$$= 7360 \text{ lb. per hour,}$$

and the production efficiency will be

$$= 12{,}189/7360$$

$$= 1{\cdot}656.$$

PRACHE AND BOUILLON EVAPORATOR

This evaporator is of French origin and is manufactured in this country by Messrs. P. & B. Evaporators, Ltd., London. The scheme of evaporation embodies the method of the compression of a portion of the vapour from the evaporator into the heating coils, by the live steam supply entering the heating coils or chamber. This method of compressing a portion of the vapour into the heating chamber is by no means new, as apparently it was first suggested in the year 1830. Various methods of compression have been tried, including piston-compressors, turbo-compressors, and steam-jet compressors. These methods did not meet with a great measure of success until Prache and Bouillon in conjunction with Sautter Harlé, engineers, Paris, studied the steam-jet compression method afresh. After many years of experimental work, they finally evolved what they termed the thermo-compressor shown in fig. 119. This thermo-compressor has an efficiency much in advance of that obtained with piston or turbo-compressors or injector compressors, and is the secret of the high production efficiency obtained with these plants.

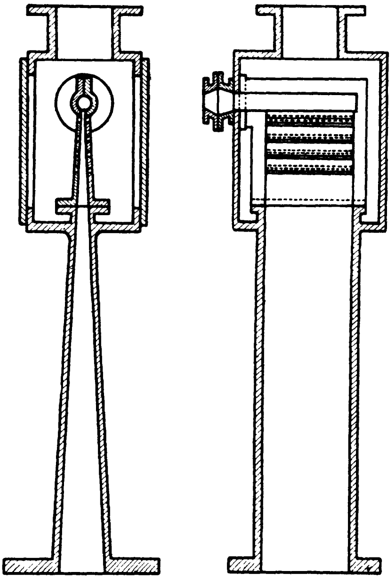

Fig. 119.—Prache & Bouillon Thermo-compressor

Production efficiency here is taken to mean the ratio between the weight of pure water made to the weight of the live steam used. The gain in efficiency is claimed to be 30 per cent over the piston compressor, and 22 per cent over the turbo-compressor. Fig. 120 shows in diagrammatic form a single effect, P. & B. evaporating plant for the production of pure water. J and H are surface heaters or heat exchangers, as they are called, and comprise a large number of small diameter straight tubes attached to tube plates and arranged inside a casing. L is a large ebullition chamber, and

N is a return pipe. P is a propeller driven by an electric motor, and functions to cause a circulation of the raw water being evaporated; the water circulates up the inside of the tubes of the heat exchanger J, where it receives heat from the steam entering from the thermo-compressor A and flows over the outside of the tubes. The heated water then enters the ebullition chamber L, where the vapour escapes upwards, and the water falls to the return pipe N to be recirculated by the propeller P. A continuous circulation therefore is set up round the heat exchanger J, ebullition chamber L, and return pipe N. The pressure and temperature in the heat exchanger J are so arranged that boiling does not take place at the heating surfaces but only heating of the water. When the water rises into the ebullition chamber, the reduction in pressure consequent on the reduced head causes ebullition, so that self evaporation takes place, and the released vapour passes upwards and out by the vapour pipe B, while the unevaporated water falls to be recirculated. By this arrangement and the fact that the speed of circulation is maintained over the critical, so that turbulent flow is maintained in the tubes, precipitation or the formation of scale does not take place inside the tubes. In cases of extremely bad or heavy scaling waters, a small quantity of abrasive material, such as sand, is circulated with the water, so that any scale which may form is removed. If the correct proportion of abrasive is used, it is claimed that no wear in the tubes takes place. The vapour leaving the ebullition chamber by the pipe B is divided, one portion passes by the pipe C to the feed heaters H, the other portion by the pipe D to the thermo-compressor A. Live steam enters the thermo-compressor by the pipe E; the live steam passes a convergent-divergent nozzle of peculiar shape, and entrains and elevates in pressure and temperature the vapour passing down the pipe D. The entrained vapour, together with the steam used in the thermo-compressor, is discharged by the latter to the heat exchanger J.

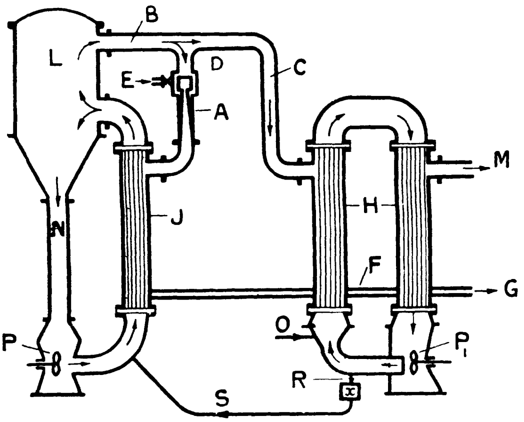

Fig. 120.—Single Effect, P. & B. Evaporating Plant

The feed heaters H are of similar construction to the heat exchanger J, that is, they comprise tube nests arranged inside a casing. The propeller P_1 creates an internal circulation as indicated by the arrows. The vapour flowing through the pipe C enters the left-hand feed heater and, flowing downwards outside of the tubes, is partially condensed. The condensate, together with the condensate from the heat exchanger J, passes across by the pipe F to the right-hand heater, and the uncondensed vapour rising upwards through the right-hand heater is condensed therein; the condensate falls to the bottom and passes out by the pipe G, together with the condensate from J and the left-hand heater. M is a vent for any surplus vapour. The raw water to be treated enters the feed heater at O, and after being heated is withdrawn at R and passes to a decanting vessel *x*, thereafter it passes to the heat exchanger J by the pipe S, and enters the general circulation in J and N.

These evaporators are designed for very low temperature differences, the difference being normally about 18° F., and the evaporation taking place at atmospheric pressure.

The approximate thermal conditions which would obtain in a simple plant, such as is shown in fig. 120, are as follows. About 1·75 lb. of vapour from D would be entrained and raised in pressure and temperature from atmospheric pressure and 212° F. to 20 lb. per square inch absolute and 228° F. respectively, for every pound of live steam used in the thermo-compressor. There would thus enter the heat exchanger J 2·75 lb. of steam at 20 lb. per square inch absolute and 228° F.; an equivalent quantity of water would be evaporated in L giving nearly 2·75 lb. 2·75 lb of vapour therefore will flow through the pipe B, 1·75 lb. passing down pipe D, and 1 lb. passing by pipe C to the feed heaters H. The total water leaving the apparatus will be 1 lb. leaving at G and coming from pipe C, plus 2·75 lb. leaving at G and coming from the heat exchanger J, or a total of 3·75 lb. Of this 3·75 lb. 1 lb. will enter the thermo-compressor at E as live steam, the total made water therefore is 3·75 − 1, or 2·75 lb. for an expenditure of 1 lb. of live steam.

The production of fresh water actually obtained in test by these evaporators per pound of steam used is:

For a simple effect	3 lb.
For a double effect	5 lb.
For a triple effect	7 lb.
For a quadruple effect	8 lb.

The temperature of the feed water in each case being 60° F.

CHAPTER X

Surface Feed Water Heaters

Surface feed water heaters may be divided into two classes: high-pressure heaters and low-pressure heaters. The high-pressure heater is so named because it is placed on the discharge side of the boiler feed pump, and is therefore subjected on its water side to at least the boiler pressure. This type of heater is generally found on board ship. The low-pressure heater is placed on the discharge side of the water-extraction pump, so that the heater is only subjected on its water side to the water-extraction pump discharge pressure, which normally is from 5 to 10 lb. per square inch for open feed water systems. This type of heater is generally found on land installations.

The main reason for the difference between the marine and land practice, for the disposition of the feed water heater, is because land power stations are universally fitted with economizers, whereas in marine practice the economizer is the exception rather than the rule. For every power plant there is a most economical feed temperature, which depends among other factors upon whether economizers are fitted or not. If no economizer be fitted, the economical feed temperature for boiler pressures between 225 and 250 lb. per square inch is about 220° F., whereas, if an economizer be fitted, the economical feed temperature is about 160° F.

With the former arrangement, i.e. no economizers, and the heater placed on the discharge of the water-extraction pump, the feed water being the discharge from the heater into the feed tank it would be at a temperature of 220° F., and would instantly boil on entering the feed tank. This would cause hammer in the feed pumps, due to vaporization in the feed pump suction, unless the feed tank be placed in an elevated position. This is not always convenient on board ship, and is decidedly bad practice in naval vessels; also owing to the steaming which will take place in the feed tank, a large loss of feed water will result. If displacement air pumps are fitted these do not work smoothly nor efficiently when discharging against a head, and the air discharged by the air pump must be released before the water is discharged to the heater. This means the fitting of a hotwell pump to deliver the water through the heater to the main feed tank. For these reasons the surface feed water heater is placed on the discharge side of the boiler feed pump, on installations which have no economizers fitted to the boilers and which have the feed system arranged on the open principle (see p. 248). On land installations with economizers these difficulties do not arise, owing to the lower feed temperature, so that the feed heater placed on the discharge side of the water-extraction pump gives good results. The low-pressure heater has a great advantage over the high-pressure, because its pressure is lower; also the heater is lighter, less costly and easier to manufacture; it is also more reliable, as with a well-made low-pressure heater troubles due to burst tubes or leaky joints are very remote. With some large modern power stations, which have high boiler pressures, the feed heating is carried out in stages, the heating steam being drawn from various expansion stages of the turbine. Some of these heaters are placed between the water-extraction pump and the feed tank according to the temperature and pressure conditions, and some between the boiler feed pump and the economizer, so that some are low-pressure heaters and some are high-pressure heaters.

NECESSARY SURFACE IN FEED WATER HEATERS

The surface of a feed water heater depends on the pressure and temperature of the steam supply; the inlet and outlet temperature of the water supply, upon the allowable heat flow across the tube surface, and the water velocity through the tube. Fig. 121 represents one tube of a surface heater.

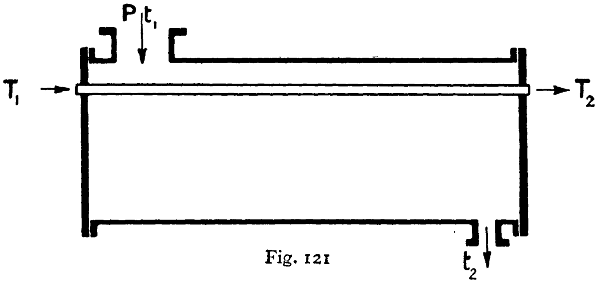

Fig. 121

Let T_1 be the inlet temperature of the feed water in degrees F.;
T_2, the outlet temperature of the feed water in degrees F.;
P, the steam supply pressure in pounds per square inch absolute;
t_1, the steam supply temperature in degrees F.;
t_2, the drain temperature in degrees F.;
S, the surface in square feet;
R, the resistance to heat flow;
L, the latent heat in the steam per pound;
q, the dryness fraction;
W, the weight of steam condensed in pounds per hour;
G, the quantity of feed passing in gallons per minute;
v, the water velocity in the tube in feet per second;
w, the weight of steam condensed in pounds per hour per square foot of surface;
W_w, the weight of water passing in pounds per hour; and
TR, the transmission rate in B.Th.U. per hour per square foot per degree F.

The resistance to heat flow, R, depends upon the mean temperature of water and its velocity, and upon the material and thickness of the tube wall. It also depends upon the degree of aeration of the entering feed water. If the feed water entering the heater be highly aerated, the heating at first proceeds slowly; this is due to the fact that the air held in solution in the water is given off very slowly up to a temperature of 176° F., when thereafter the air is given off freely. The air liberated at temperatures below 176° F. seems to have the property of adhering to the tube surface, thus forming a thin film which acts as an insulator and prevents the free flow of heat from the tube wall to the water. At temperatures above 176° F. the freely liberated air is easily brushed off the interior of the tube, by the rapidly moving water, and the heat flow takes place readily. This can be roughly demonstrated if a beaker be filled with water, highly aerated, drawn from a household tap. If the beaker be placed over a bunsen flame and a thermometer suspended in the water, it will be noticed that as the temperature rises to 176° F., the water remains quite clear; at 176° F. the appearance of the water suddenly changes and becomes quite milky, indicating the sudden liberation of the air; this milky appearance gradually disappears as the temperature of the water approaches boiling-point, showing that the liberation at boiling-point is complete.

For surface feed water heaters heating the condensate from a turbine, the aeration of the condensate is not sufficiently high for the above phenomenon to be of any practical importance and may be neglected.

Assuming this to be the case, then the laws pertaining to the heat flow in surface feed water heaters will be similar to those given for steam condensers (see pp. 7 to 10).

The resistance to heat flow is given by

$$R = (1000 \times \theta_m)/(wqL) \ldots\ldots\ldots\ldots\ldots (125)$$

As before, R is made up of the three resistances, R_a, R_t, and R_w.

$$\text{i.e. } R = R_a + R_t + R_w.$$

Many heaters of low-pressure design are fitted with ordinary condenser tubes, and in such cases the numerical values of the above resistances will be the same as those given for steam condensers. For high-pressure heaters the tubes are usually of copper having a wall thickness about 10 S.W.G., and for such heaters the numerical values of the resistances are different from those already given.

The value of R_t can be found as follows:

The absolute conductivity of copper may be taken as 0·00285 B.Th.U. per second per 1 in. cube per degree Fahrenheit difference.

Now we know that

$$R_t = \frac{\text{temperature difference}}{\text{heat flow}},$$

and, as before, increasing the unit of resistance 1000 times we get for tubes $\frac{1}{8}$ in. thick

$$R_t = \frac{1 \times 1000}{0{\cdot}00285 \times 144 \times 3600 \times 8}$$
$$= 0{\cdot}084.$$

The numerical values of R_w may be found from the equation

$$R_w = C/v^{0{\cdot}82},$$

where C is a constant found from fig. 122 at the average temperature of the feed water passing through the heater. The numerical value of R_a depends, as in steam condensers, on the proper air drainage arrangements and on the conditions under which the heater is operating; if the heater is supplied with exhaust steam which contains oil or grease from reciprocating auxiliaries, this is deposited on the heating surface and impairs the heat flow, and an allowance should be made for this. If the water passing through the tubes contains salts which are likely to be deposited on the tubes, an allowance should also be made for this. Generally R_a varies from 0·5 to 1·1.

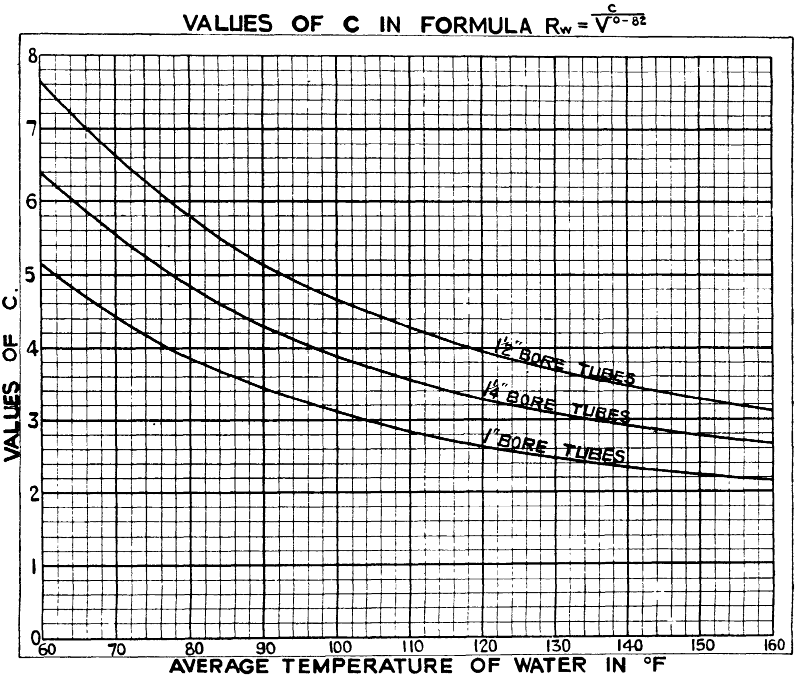

Fig. 122.—Values of C

The action going on in a surface feed heater is that of a heat exchange so that the heat given up by the steam is equal to the heat taken up by the water, therefore by equation (16), see p. 16, we have

$$G = \{W(qL + t_1 - t_2)\} / \{600\,(T_2 - T_1)\} \text{ gall. per minute.}$$

Example 1.—A surface feed water heater is to raise 100,000 lb. of feed water per hour from 80° to 140° F. when supplied with steam at an absolute pressure of 20 lb. per square inch. The temperature of the drain from the steam space is 200° F. Calculate the quantity of steam necessary, allowing 5 per cent of the heat supplied as lost in radiation. The dryness fraction of the steam is 0·95.

From equation (16) and taking 95 per cent of the heat supplied as being absorbed by the feed water we have

$$\begin{aligned} 100{,}000/60 &= [W\{(0{\cdot}95 \times 960{\cdot}5) + (228 - 200)\}] \,/\, [600(140 - 80)0{\cdot}95, \\ \text{or}\quad W &= 100{,}000 \times 60 \times 0{\cdot}95/940{\cdot}5 \\ &= 6060 \text{ lb. per hour.} \end{aligned}$$

Example 2.—A surface feed water heater is to heat 75,000 lb. of feed water from 75° to 200° F. when supplied with exhaust steam at a pressure of 5 lb. per square inch gauge, having a total heat content of 1100 B.Th.U. per pound reckoned from 32° F. Assuming that the drain water leaves the heater at steam temperature, calculate the surface necessary. The tubes are of copper $1\frac{1}{4}$ in. bore, 10 S.W.G. thick, and the water velocity is 5 ft. per second.

The total heat content is 1100 B.Th.U. per pound reckoned from 32° F., therefore the heat available for feed heating per pound of steam in condensing and draining off at the temperature corresponding to 5 lb. per square inch gauge or 228° F. is

$$\begin{aligned} \text{heat available} &= 1100 - (228 - 32) \\ &= 904 \text{ B.Th.U.} \\ \text{or}\quad qL &= 904. \end{aligned}$$

$$\text{Now } R = R_a + R_t + R_w.$$

$$\begin{aligned} \text{Taking } R_a &= 0{\cdot}8 \\ \text{and}\quad R_t &= 0{\cdot}084 \\ \text{and}\quad R_w &= C/v^{0\,82}. \end{aligned}$$

From fig. 122, C = 2·96 for the average temperature of

$$\begin{aligned} (75 + 200)/2 &= 137{\cdot}5^\circ \text{ F.} \\ \text{and}\quad v^{0{\cdot}82} &= \text{antilog of } (0{\cdot}82 \log 5) \\ &= \text{antilog of } 0{\cdot}82 \times 0{\cdot}699 \\ &= \text{antilog of } 0{\cdot}573 \\ &= 3{\cdot}743. \\ \therefore\ R_w &= 2{\cdot}96/3{\cdot}743 = 0{\cdot}791. \\ \therefore\ R &= 0{\cdot}8 + 0{\cdot}084 + 0{\cdot}791 \\ &= 1{\cdot}675. \end{aligned}$$

The transmission rate TR will therefore be

$$\begin{aligned} \mathrm{TR} &= q\mathrm{L}/\mathrm{R} \\ &= 904/1{\cdot}675 \\ &= 539{\cdot}7 \text{ B.Th.U. per hour per square foot per degree F.} \end{aligned}$$

From equation (8), p. 4, the mean temperature difference is

$$\begin{aligned} \theta_m &= (\theta_1 - \theta_2) / \{2{\cdot}3 \log_{10}\theta_1/\theta_2\} \\ \theta_1 &= 228 - 75 \\ &= 153 \\ \theta_2 &= 228 - 200 \\ &= 28. \\ \text{Hence} \quad \theta_m &= 125/\{2{\cdot}3 \log_{10} 153/28\} \\ &= 125/2{\cdot}3 \times 0{\cdot}738 \\ &= 125/1{\cdot}696 \\ &= 73{\cdot}7^\circ \text{ F.} \end{aligned}$$

The surface necessary will therefore be

$$\begin{aligned} \mathrm{S} &= 75{,}000\,(200 - 75)/73{\cdot}7 \times 539{\cdot}7 \\ &= 236{\cdot}2 \text{ sq. ft.} \end{aligned}$$

Example 3.—A surface feed water heater, having 300 sq. ft. surface and fitted on a passenger mail boat, raised 120,000 lb. of feed water from 90° F. to 220° F., when the steam pressure in the shell was 25 lb. per square inch gauge and the temperature of the drain 230° F. The heater was situated on the discharge side of the boiler feed pump, and was fitted with $1\frac{1}{2}$-in. bore copper tubes, 10 S.W.G. thick. Calculate the heat flow across the tube surface per degree F. of mean temperature difference per hour per square foot, and hence deduce the sum of the resistances $(\mathrm{R}_a + \mathrm{R}_t)$. The tube velocity was 8 ft. per second.

The steam temperature corresponding to 25 lb. per square inch gauge is 267° F.

The mean temperature is

$$\begin{aligned} \theta_m &= (\theta_1 - \theta_2)/\{2{\cdot}3 \log_{10}\theta_1/\theta_2\} \\ &= \{(230 - 90) - (267 - 220)\}/\{2{\cdot}3 \log_{10}(230 - 90)/267 - 220\} \\ &= 93/(2{\cdot}3 \times 0{\cdot}474) \\ &= 85{\cdot}3^\circ \text{ F.} \end{aligned}$$

The heat flow per square foot of surface per degree F. of mean temperature difference per hour

$$\begin{aligned} &= \text{total heat flow}/(\theta_m \times \text{surface}) \\ &= 120{,}000(220 - 90)/(85{\cdot}3 \times 300) \\ &= 609 \text{ B.Th.U.} \end{aligned}$$

Again $\quad \mathrm{R}_a + \mathrm{R}_t + \mathrm{R}_w = 1000 \times \theta_m/(wq\mathrm{L}),$

or $\quad (\mathrm{R}_a + \mathrm{R}_t + \mathrm{R}_w)/1000 = \theta_m/(wq\mathrm{L}).$

Inverting this equation we get

$$1000/(R_a + R_t + R_w) = wqL/\theta_m,$$
$$\text{or} \quad 1000/(R_a + R_t + R_w) = WqL/(\theta_m S).$$

The right-hand side of this equation is the heat flow per square foot of surface per degree F. of mean temperature difference per hour.

$$\therefore\ 1000/R_a + R_t + R_w) = 609.$$
$$\begin{aligned} \text{Now} \quad R_w &= C/v^{0\cdot 82} \\ &= 3\cdot2/8^{0\cdot82} \\ &= 3\cdot2/5\cdot5 \\ &= 0\cdot582. \end{aligned}$$

Then, substituting this value of R_w in the above equation, we get

$$\begin{aligned} 1000/(R_a + R_t + 0\cdot582) &= 609, \\ \text{or} \quad R_a + R_t &= (1000/609) - 0\cdot582 \\ &= 1\cdot642 - 0\cdot582 \\ &= 1\cdot06. \end{aligned}$$

SURFACE FEED HEATERS FITTED WITH STEEL TUBES

For feed heaters fitted with steel tubes the resistance to heat flow across the tube wall R_t is greatly increased, and for equal duties the steel-tube heater is much larger than the copper-tube heater.

The following values of R_t may be taken: 3·5 for tubes 10 S.W.G. thick, and 1·75 for tubes 16 S.W.G. thick.

SUPERHEATED STEAM IN SURFACE FEED HEATERS

When superheated steam is supplied to surface heaters, the film of moisture which deposits on the steam side of the tube is at saturation temperature, otherwise the film could not exist as water. The temperature gradient across the tube wall will therefore be the same as if saturated steam had been supplied, and consequently the heat flow in unit time will be the same for the two qualities of steam. The surface necessary therefore, for a given heat flow, is the same whether the supply steam be superheated or saturated, and this is borne out in actual practice. The surface should be fixed up by calculating the mean temperature difference for saturation temperature, and this value used in the equation deduced for saturated steam.

The weight of superheated steam necessary to give a given temperature rise to the feed water should be found by taking the superheat into account. Thus if t_s be the steam temperature, and γ the specific heat of the steam, then the heat given up by the steam is

$$W\{\gamma(t_s - t_1) + L + t_1 - t_2\},$$

and the heat absorbed by G gall. of feed water per minute is

$$600G(T_2 - T_1).$$

Equating these equal quantities of heat we get

$$W[\gamma(t_s - t_1) + L + t_1 - t_2] = 600G(T_2 - T_1),$$

or $$W = \{600G(T_2 - T_1)\}/\{\gamma(t_s - t_1) + L + t_1 - t_2\} \quad \ldots.(126)$$

Example.—A stage feed water heater in a 30,000-kw. turbo-generator set is supplied with steam extracted from a stage of the main turbine, where the steam pressure is 45 lb. per square inch absolute, and the steam temperature is 310° F. The quantity of feed water passing through the heater is 290,000 lb. per hour, and it enters at a temperature of 150° F. Assuming that the steam is just condensed in the heater, calculate the quantity of steam which must be extracted from the turbine to give an outlet feed temperature from the heater of 250° F. and find the surface necessary. Take the specific heat of the steam as 0·55. The tubes are of copper $1\frac{1}{4}$ in. bore and 10 S.W.G. thick, and the tube velocity is 7 ft. per second.

The mean temperature is found from equation (8) (p. 4).

$$\begin{aligned}\theta_m &= (\theta_1 - \theta_2)/\{2{\cdot}3 \log_{10}\theta_1/\theta_2\}\\ &= \{(274 - 150) - (274 - 250)\}/\{2{\cdot}3 \log_{10}(274 - 150)/(274 - 250)\}\\ &= 100/(2{\cdot}3 \times 0{\cdot}714)\\ &= 60{\cdot}8^\circ \text{ F.}\end{aligned}$$

From equation (126)

$$\begin{aligned}W &= 600G(T_2 - T_1)/\{\gamma(t_s - t_1) + L + t_1 - t_2\}\\ &= 290{,}000(250 - 150)/\{0{\cdot}55(310 - 274) + 928 + 274 - 274\}\\ &= 290{,}000 \times 100/947{\cdot}8\\ &= 30{,}500 \text{ lb. per hour.}\end{aligned}$$

From equation (125) we have

$$R = 1000 \times \theta_m/(wqL).$$

Now qL is the heat flow per pound of steam condensed, which for superheated steam is

$$\gamma(t_s - t_1) + L + t_1 - t_2,$$

then, substituting this value for qL, equation (125) becomes

$$R = 1000 \times \theta_m/[w\{\gamma(t_s - t_1) + L + t_1 - t_2\}]. \quad \ldots..(127)$$

For copper tubes, 10 S.W.G. thick, $R_t = 0{\cdot}084$, and $R_a = 0{\cdot}8$.

To find R_w we have the average temperature of the feed water $(150 + 250)/2 = 200^\circ$ F., and, from fig. 122, C for this temperature is 2·5. (Found by extending the curve for $1\frac{1}{4}$-in. bore tubes.)

Then $$\begin{aligned}R_w &= C/v^{0{\cdot}82}\\ &= 2{\cdot}5/7^{0{\cdot}82}\\ &= 2{\cdot}5/4{\cdot}92\\ &= 0{\cdot}508.\end{aligned}$$

$$\begin{aligned} \text{Now} \quad R &= R_a + R_t + R_w \\ &= 0{\cdot}8 + 0{\cdot}084 + 0{\cdot}508 \\ &= 1{\cdot}392. \end{aligned}$$

Substituting these values in equation (127) we get

$$\begin{aligned} 1{\cdot}392 &= 1000 \times 60{\cdot}8/[w\{0{\cdot}55(310 - 274) + 928 + 274 - 274\}], \\ \text{or} \quad w &= 1000 \times 60{\cdot}8/(1{\cdot}392 \times 947{\cdot}8) \\ &= 46{\cdot}1 \text{ lb. per square foot.} \end{aligned}$$

The surface then will be

$$\begin{aligned} &= 30{,}500/46{\cdot}1 \\ &= 661{\cdot}6 \text{ sq. ft.} \end{aligned}$$

DIRECT-CONTACT FEED HEATERS

The direct-contact feed water heater performs its function of heating feed water by the direct contact of the feed water with the heating steam. The action in these heaters is similar to that in a jet condenser, in that the feed water is sprayed into the heater so that complete co-mingling of the water and steam takes place.

This type of heater has many advantages over the surface type, the chief one being that the temperature and pressure conditions can be so arranged that the feed water leaving the heater is de-aerated, and is thus rendered ideal for boiler feed purposes.

The relation between the temperatures and quantities in this type of heater is the same as that deduced for jet condensers. (See equation (37), p. 78.)

Fig. 123 shows a typical example of this type of heater, which has been applied to many ships of the mercantile marine. The heater illustrated is of Messrs. G. & J. Weir's manufacture.

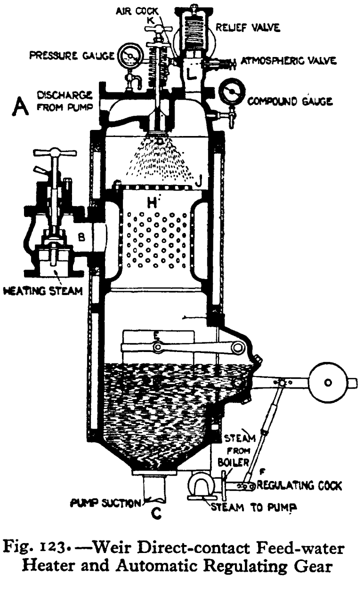

Fig. 123.—Weir Direct-contact Feed-water Heater and Automatic Regulating Gear

The heater is situated high up in the engine-room, and the feed water is pumped from the hotwell up to the heater inlet branch A; the water leaves the heater by the branch C, and flows direct to the feed pump suction. The water entering at A passes thereafter through the spring-loaded spray valve D. The steam enters through the non-return, steam inlet valve B, and thereafter passes through the holes in the waist piece H and meets the feed water raining down from the holes in the plate J. The steam and feed water are thus well mixed, and the mass of feed water and condensed steam falls to the bottom of the heater. E is an open float pan which operates a steam regulating valve F; this regulating valve controls the steam to the feed pump, which is drawing from the heater, so that the water level in the heater is maintained constant under varying quantities of feed water being dealt with. The sudden release of pressure on the feed water passing through the spray valve, combined with the heating of the water, has the effect of releasing the air held in solution by the feed water. A special air collecting dome L is fitted with a cock K. A small pipe is led from this cock to the main condenser; the air which is released in the heater is thereby withdrawn to the main condenser, so that the temperature of the water leaving the heater is maintained at the temperature corresponding to the absolute pressure in the heater. This condition of temperature with corresponding pressure renders the water leaving the heater air free, and non-corrosive. (See pp. 249 and 250.)

Since the water leaving the heater is hot, the heater must be placed at such an elevation that knocking in the feed pumps, if these be of the direct-acting type, is to be prevented, and if of the rotary type cutting out is to be prevented. The float E with regulating valve F ensures that the static head on the feed pump suction is maintained under all conditions of load, so that troubles due to knocking in the feed pumps are obviated. For a feed temperature of 200° to 220° F. these heaters are given an elevation of about 40 ft.

These heaters have been fitted to ships of the mercantile marine for the past thirty years; it is, however, only within recent years that their application to land installations has been adopted. This has been primarily brought about by the advent of the steel-tube economizer, which is particularly susceptible to corrosion caused by dissolved air or oxygen in the feed water.

CHAPTER XI

Feed Systems

The feed system in any power plant is usually reckoned to be the system through which the water goes from the time it leaves the condenser as condensate till the time it arrives at the economizer or boiler as feed water. There are three main systems in use: first, the open system; second, the closed system; and, third, a combination of both.

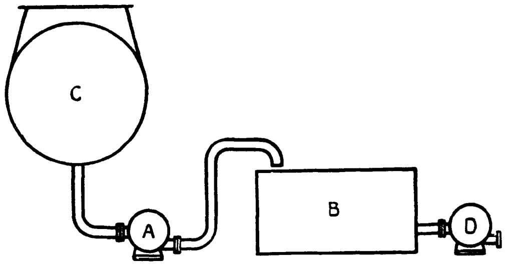

Fig. 124.—Open Feed System

Fig. 124 shows, in diagrammatic form, the open system, which until recent years was the one usually adopted.

The water-extraction pump A withdraws the water of condensation from the condenser C, and delivers it into the open feed tank B. The boiler feed pump D draws from the feed tank and delivers direct to the boilers. The feed tank B is open to the atmosphere, and the system derives its name from this fact.

Fig. 125 shows, in diagrammatic form, the closed-feed system.

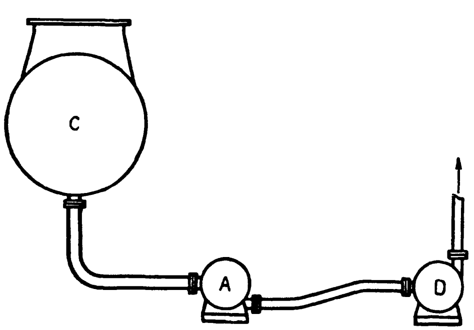

Fig. 125.—Closed Feed System

The water-extraction pump A withdraws the water of condensation from the condenser C, and discharges direct to the suction of the boiler feed pump D, which discharges direct to the economizers or boilers. It will be noticed that the condensate or feed water in this arrangement does not come into contact with the atmosphere, and is thus given no opportunity to absorb air. The water flows through a closed feed line, and this fact gives the name " closed " to the system.

The problem of corrosion in the boilers, economizers, and feed pipes, &c., has been receiving much attention within recent years, and many theories have been advanced to account for the action.

There is no one theory which is generally accepted to-day. Two broad theories have, however, developed, and opinions are divided as to which is the true one. The older or the so-called acid theory was summarized by Calvert in 1871. According to this theory oxygen and pure water cannot by themselves induce corrosion, the presence of an additional substance being necessary. This additional substance is taken to be carbonic acid, which is generally present when oxygen is present. The action may be summarized as follows:

Carbonic acid in solution in water in the presence of iron is decomposed to ferrous carbonate, and nascent hydrogen is produced. The ferrous carbonate is in turn decomposed and carbonic acid is liberated, and is thus free to attack more iron. The carbonic acid thus acts in the nature of a catalyst, and only a small quantity of carbonic acid is necessary to produce active corrosion.

There are other substances which act catalytically, like carbonic acid, and promote corrosion. Sodium chloride, even when rendered slightly alkaline, will promote or stimulate the corrosion.

The newer or modern view of corrosion was first suggested in 1903 by Whitney, and may be termed the electrolytic theory of corrosion. Whitney suggested that since water is an electrolyte it is split up to a small extent into hydrogen ions and hydroxyl ions, and that the presence of another substance to act as a catalyst is quite unnecessary. According to this theory some of the iron passes into solution in the ionic condition, and an equivalent amount of hydrogen is liberated at the surface of the metal. This liberated hydrogen is depolarized by the oxygen in solution in the water and more iron passes into solution. The action goes on till saturation is reached, when hydrated ferrous oxide, more or less oxidized to the ferric condition, precipitates out. A continual supply of oxygen therefore is necessary for the action to continue.

Chemists tell us that the modern electrolytic theory is not sufficient to account for all the phenomena observed, and a modified or auxiliary theory has been brought forward by Dr. J. Newton Friend. According to this theory the slowly oxidized ferrous hydroxide is formed in the colloidal condition, which in the presence of air is oxidized to a higher oxide, still in the colloidal condition. This higher colloidal hydroxide catalytically accelerates the action, and when the colloid precipitates out, ordinary rust is formed. (See *The Journal of the West of Scotland Iron and Steel Institute*, February, 1924.)

However near these theories may be to the truth, it is now generally accepted by engineers that, if the feed water used be pure distilled water, the corrosion, which is found to take place in the feed system, &c., is due to dissolved gases in the feed water, and all modern feed systems are designed with a view to the elimination of these gases. The dissolved gases found in the feed water are those of which the air is composed, namely, nitrogen, oxygen, and small percentages of carbon dioxide and free hydrogen. The gases responsible for the corrosion are the oxygen and the carbon dioxide;

nitrogen, being an inert gas, plays no part in the corrosion. The absorption of these gases by water obeys Henry's law for the solubility of gases. Henry's law states that " the volume of a gas absorbed by a liquid is directly proportional to the pressure of the gas ". This means that, if the pressure of the gas be halved, the volume absorbed will be halved, the volume in each case being measured at 32° F. and 30 in. of mercury pressure. The coefficient of absorption first introduced by Bunsen is the volume of the gas measured at 0° C. and 760 mm. which is absorbed by 1 c.c. of the liquid. For mixed gases, such as air, the solubility of the gases is proportional to their partial pressures, that is, if the partial pressure be reduced to zero, the solubility is reduced to zero and the liquid absorbs no gas. The closed feed system then depends for its efficiency, as regards the supplying of a gas-free feed water to the boilers, on the partial air pressure which obtains at the condensate outlet in the condenser. In other words, the condensate leaving a steam condenser only contains absorbed air in proportion to the partial air pressure which obtains at the condensate outlet in the condenser. With good condensers, such as the regenerative condenser described on pp. 72 to 74, which delivers the condensate at practically vacuum temperature, the air held in solution is sufficiently far reduced in quantity as to make the condensate quite suitable for passing through a closed feed system direct to the boilers, with an assurance of complete protection against active corrosion due to dissolved gases.

The foregoing remarks give the essentials which led to the closed feed system apart from de-aeration plants. De-aeration plants are representative of the third system mentioned on p. 248, and will be referred to hereafter.

Reverting to fig. 125 it will be realized that we have here two pumps, A and D, working in series and tied together hydraulically, so that the pump D can only deliver the quantity supplied to it by the pump A. If the pump A tends to deliver more water than is demanded by the pump D, the water will accumulate and rise up into the condenser C, thus causing flooding in the condenser C. If, on the other hand, the pump A delivers less than is demanded by the pump D, the pressure in the pipe between the pumps will fall, and the pump D will run under a condition of cavitation which will cause it to work erratically; and, if D be turbine driven, its speed would increase sufficiently for the trip governor to act and shut the steam valve, thus stopping the pump; while, if D be a direct-acting pump, severe hammering would take place. A degree of flexibility must therefore be provided between pumps A and D, such that if A delivers more than is demanded by D the excess water will pass to the feed tank; while if A tends to deliver less water than is demanded by D, the deficiency will be drawn from the feed tank to the suction of A, and thereby enable it to deliver the quantity demanded by D. Various methods are adopted to allow for this flexibility to give a practical and smooth working to such systems.

WEIR CLOSED FEED SYSTEM

The method adopted by Messrs. G. & J. Weir, Ltd., is shown diagrammatically in fig. 126.

Two float-controlled valves E and F are provided, such that when the pump A is delivering more water than is demanded by the feed pump D, the valve F automatically opens and the excess water passes to the feed tank G, thus ensuring no flooding in the condenser C, and when the pump A tends to deliver less water than the pump D demands, the valve E opens and water is drawn up from the feed tank G into the condenser, thus making good the deficiency and enabling the pump A to supply the quantity demanded by D. The automatic operation of the valves F and E is performed by the hollow copper floats H and J; these floats are housed in a float box, which is connected top and bottom to the condenser C and the suction pipe of the pump A respectively, so that the water level in the float box is the same as that in the suction pipe K. These floats are set at different levels, and operate small pilot or leak-off valves, so that the opening or closing of these pilot valves opens or closes hydraulically the valves E and F. The top float H controls the valve F, which opens when the float rises; the bottom float J controls the valve E, which opens when the float falls. The automatic working of the system may now be understood. If the pump A tends to deliver more water than is being demanded by the pump D, the water level in the suction pipe K will rise, thus raising the float H; the pilot valve under the control of H will close, and the water pressure will build up along the small pipe L, which, being connected to

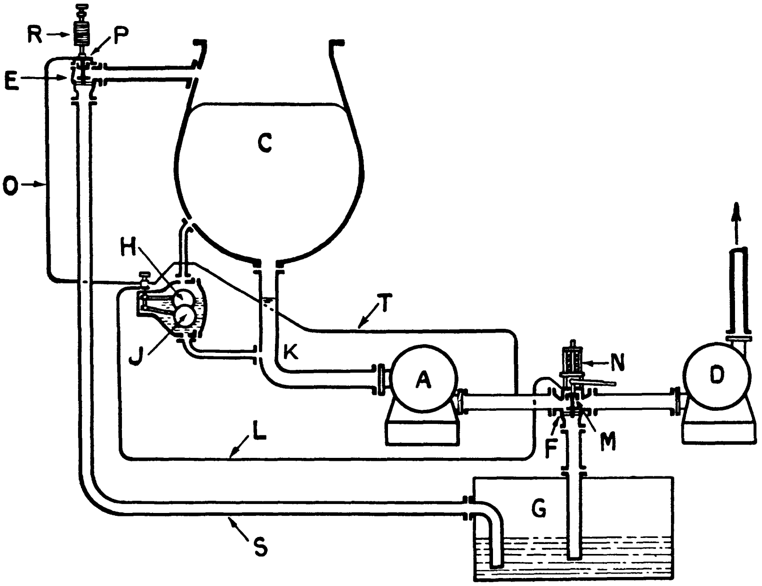

Fig. 126.—Weir Closed Feed System

the valve F, allows the pressure to build up on the top of the piston M, thus opening the valve against the spring N and allowing the excess water to pass to the feed tank G. Again, if the pump A tends to deliver less water than is demanded by the pump D, the water level in the suction pipe K will fall and the float J will also fall, thus opening the pilot valve under the control of this float. The float H will have meantime fallen and closed the valve F. The pressure along the control pipe O will consequently fall and relieve the water pressure on the top of the piston P, and allow the spring R to open the valve E. The vacuum in the condenser will then draw water from the feed tank G through the pipe S and into the top of the condenser. This water falls down through the condenser, and becomes de-aerated by the influence of the vacuum, and flows to the suction of the pump A, thus enabling A to deliver to D the quantity of water demanded by the latter. The operating water for these valves is taken from the discharge of the water-extraction pump A; the operating water for the valve F enters the control cylinder by a small hole drilled through the piston M, and the operating water for the valve E is led by the pipe T to the pilot valve under the control of the float J, and thence to the control cylinder of the valve E. Fig. 127 shows a section through the

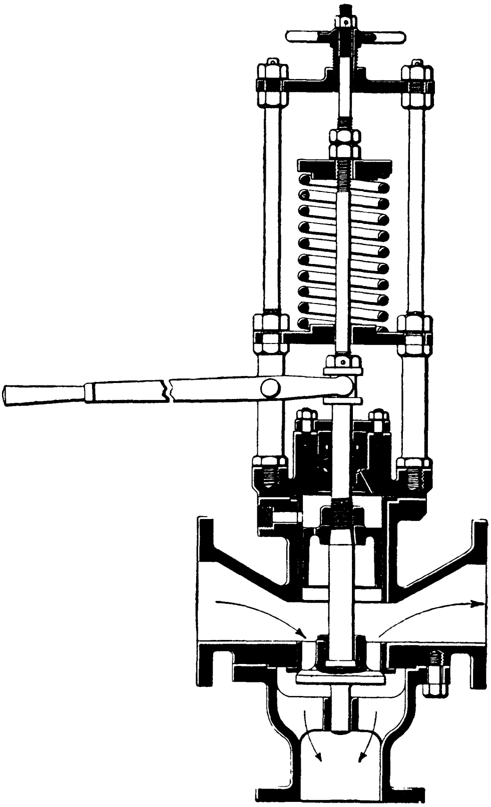

Fig. 127

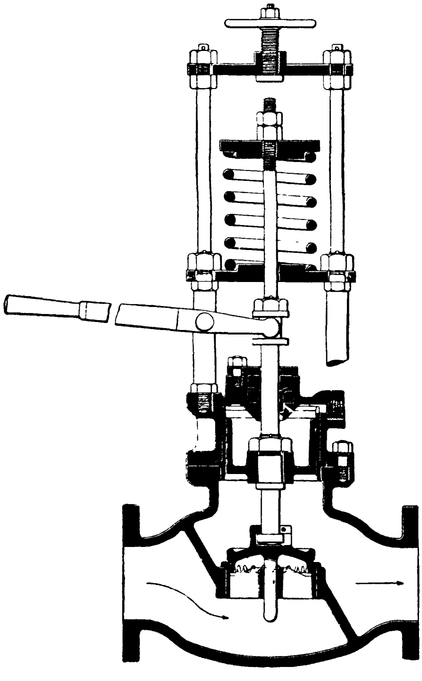

Fig. 128

valve F; fig. 128 shows a section through the valve E; and fig. 129 shows a section through the float control box.

The above description gives the methods adopted to obtain the necessary flexibility between the pumps A and D; the difference in levels at which the floats H and J are set is normally 6 in., which difference has been found to give satisfactory working. With the above arrangement it should be noted that all make-up introduced into the system first passes through the main condenser, where it is de-aerated to the extent of the conditions prevailing in the condenser before the water is passed to the feed system.

As adopted for ships of the mercantile marine the whole scheme of ejectors, feed heaters, &c., is shown in fig. 130. The float-controlled, automatic valves are shown at J and K; the float box is shown at R; B is the water-extraction pump; and H the main boiler feed pump; while H_1 is the auxiliary boiler feed pump which is generally of the direct-acting, steam-driven type. C is a drain cooler placed on the feed suction pipe between the water-extraction pump discharge and the boiler feed pump suction.

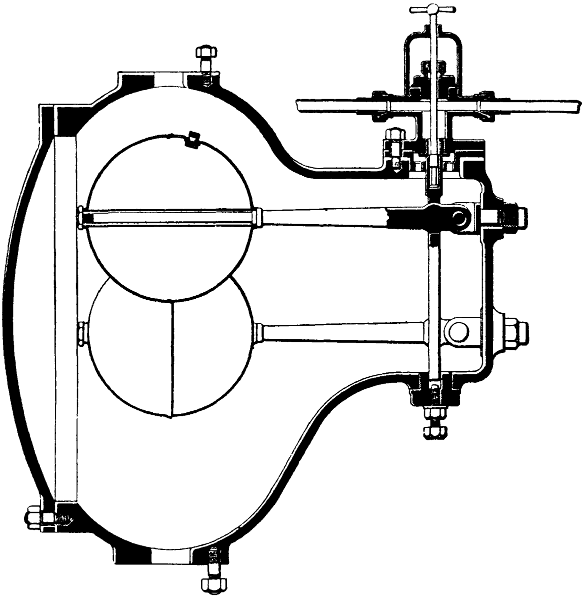

Fig. 129

The drain cooler is of the tubular type, the condensate discharged by the water-extraction pump passing through the inside of the tubes; while the drains from the main surface feed heater G and the after condenser of the air ejector E enter the underside of the drain cooler and flow upwards and in four passes to the top. The residual heat in these drains is therefore absorbed by the condensate passing through the cooler. The drains leave the cooler by the branch S and pass to the sea-water circulated condenser D, where they are further cooled and thus rendered suitable for efficient filtration in the filter L. The drain from the surface heater G usually con-

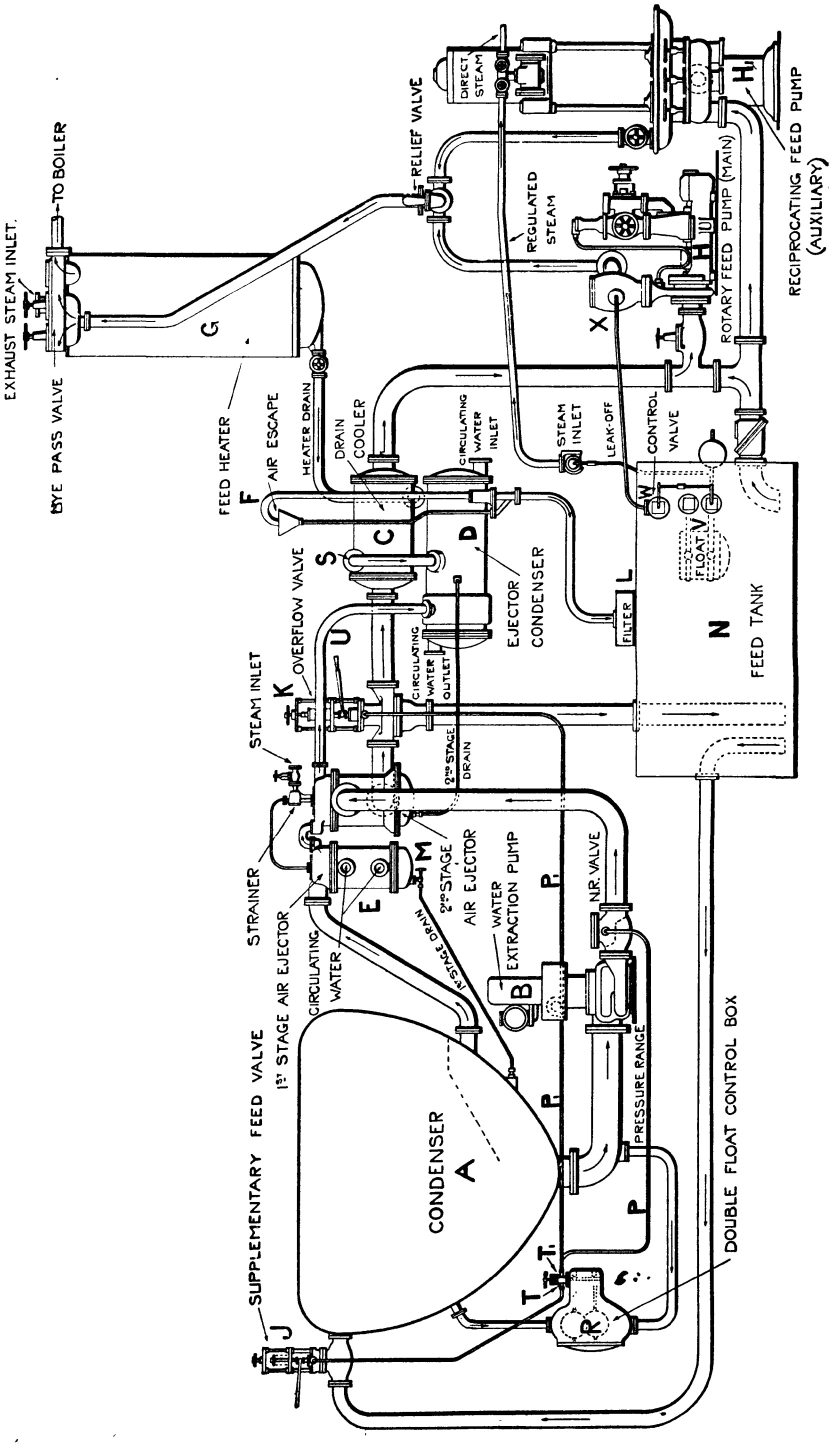
TO BOILER
EXHAUST STEAM INLET.
BYE PASS VALVE
G
FEED HEATER
F
AIR ESCAPE
HEATER DRAIN
DRAIN COOLER
C
S
OVERFLOW VALVE
K
STEAM INLET
STRAINER
1ST STAGE AIR EJECTOR
CIRCULATING WATER
E
M
2ND STAGE AIR EJECTOR
1ST STAGE DRAIN
2ND STAGE DRAIN
U
CIRCULATING WATER INLET
CIRCULATING WATER OUTLET
D
EJECTOR CONDENSER
WATER EXTRACTION PUMP
B
P
N.R. VALVE
PRESSURE RANGE
CONDENSER
A
SUPPLEMENTARY FEED VALVE
J
T
T₁
R
DOUBLE FLOAT CONTROL BOX
FILTER
L
N
FEED TANK
FLOAT
V
W
CONTROL VALVE
LEAK-OFF
STEAM INLET
X
REGULATED STEAM
RELIEF VALVE
DIRECT STEAM
H
U
ROTARY FEED PUMP (MAIN)
H₁
RECIPROCATING FEED PUMP (AUXILIARY)

Fig. 130

tains a large percentage of grease, due to the exhaust which enters the heater being exhaust from direct-acting or reciprocating auxiliary engines, which require oil for their cylinder lubrication. This drain must be properly cooled if the grease is to be efficiently extracted in the filter. The foregoing method is that adopted by Messrs. G. & J. Weir, Ltd. The condensate discharged by the water-extraction pump first passes through the after cooler of the air ejector E and absorbs the heat in the discharge from the second stage of the air ejector, the condensate then passes to the drain cooler. The inter-condenser, or the condenser which condenses the steam used in the first stage of the air ejector, is circulated with cold sea water taken from discharge of the main circulating pumps, the heat in the discharge steam from the first stage is therefore lost in the circulating water; this, however, is a small percentage of the total, being roughly 30 per cent of the heat used by the ejector. The arrangement is, however, necessary for marine duty, because the vacuum must be maintained when no condensate is being discharged by the water-extraction pump, a condition which obtains when the main turbines are stopped. It is very essential for the manœuvrability of a ship that the full vacuum be maintained in the main condensers during manœuvring, and the foregoing arrangement fulfils this essential. When the main turbines are stopped, no condensate is discharged through the after cooler, condensation in this cooler therefore ceases, and the steam discharged by the second stage ejector blows through the cooler and leaves by the discharge pipe U, which leads to the sea circulated condenser D. The steam is condensed and cooled therein, and thereafter passes to the feed tank N. Under running conditions only air and vapour pass along the pipe U from the air ejector, and enter the condenser D; the air is here devaporized and cooled, and escapes by the air escape pipe F.

The arrangement shown in fig. 130 is that adopted when the system has to operate in conjunction with Scotch boilers, and the control of the boiler feed pumps is from the water level in the feed tank. Inside the feed tank a float V is fitted which operates a control valve W; this control valve W is hydraulically connected to a special non-return valve X on the main rotary feed pump discharge to the boilers. When the float falls the control valve W operates and progressively closes the discharge valve X.

When the float rises the reverse action takes place, and the valve X is opened; the quantity of water fed to the boilers is thus automatically regulated by the water level in the feed tank. When the ship is under way the usual small supplementary feed supply from the reserve fresh-water tanks is opened to the main condenser and the losses made up thereby. The regulation of the auxiliary, or stand-by, direct-acting feed pump is obtained from the same float, the float in this case regulating a steam throttle cock which controls the steam supply to the pump; the throttle cock opens when the float rises, thus increasing the speed of the pump and vice versa when the float falls. This pump and control are only used when getting under way, or if the main feed pump breaks down.

When this system works in conjunction with water tube boilers, the above control is omitted; the feed pumps are controlled by the usual boiler water-level controllers in the boilers, and the water-extraction pump is of such a capacity that it has the ability to follow up the feed pump under all conditions. In ships fitted with water tube boilers, and especially fast naval vessels, a sudden shut down of the main engines causes a sudden and large drop in the water level in the boilers. This is due to sudden stoppage of the steaming in the boilers caused by the rise in boiler pressure and the closing of the burners. A boiler which is steaming heavily has a water level much higher than the solid water level which would obtain if no ebullition were taking place inside, or the average density of the water inside the boiler is, due to the ebullition, much less than that of solid water, hence the water level rises and is higher than that shown in the water-level gauge glass. When the steaming ceases the water level settles to approximately that for solid water, and the water-level controllers open up the feed-regulating valves; this causes the boiler feed pump to discharge its maximum capacity, and the water-extraction pump must be able to follow up the boiler feed pump. This condition calls for a very large supplementary feed valve J (fig. 130), which must allow the maximum capacity of the boiler feed pump to be drawn into the main condenser, and hence pass to the water-extraction pump, so that the latter is able to supply the demand made by the feed pump; the tank also must be of such a capacity that it is not drained empty by this condition. The boiler feed pump continues to discharge its maximum capacity till the water level in the boilers rises and operates the controllers, thus closing the regulating valves. When the main engines start away again ebullition in the boilers will commence, and the water level will rise up over the water-level controllers, in other words, there will be too much water in the boilers. This water is got rid of by the water-level controllers holding the regulating valves shut so that the feed pump discharges no water, the water level in the extraction pump suction pipe will therefore rise, and by the action of the float control open the overflow valve K, and the excess water will pass to the feed tank N. This release of water will continue till the water level in the boiler falls to normal, when the controllers will open the regulating valves and enable the feed pump to commence feeding normally. This arrangement has been found to work well on fast naval vessels of the torpedo-boat class.

MIRRLEES WATSON CLOSED FEED SYSTEM

The closed feed system as placed on the market by Messrs. The Mirrlees Watson Company, is shown in fig. 131, and which, along with the following description, was kindly supplied to the author by the above company. The condensate from the main condenser A drains to the condensate collecting tank B under condenser vacuum, and in which is housed a float ball K and valve L. The condensate pump C draws from the tank B and delivers to a pressure tank D. The pressure in this tank is regulated by the spring-loaded control valve J, and is such as may be found necessary to deliver the condensate through the feed heaters to the feed tank or feed pump. In order that the condensate pump may operate steadily under varying steam load and to provide for the accurate measurement of the steam load on the main turbine at all times, an equalizing or balance pipe M connects the pressure tank D with the condensate collecting tank B. This connection, in combination with the pressure tank and the float-control valve L, ensures the desired results. The condensate pump C is designed to deal with the condensate resulting from full steam load, under which condition a certain water level will be maintained in the collecting tank B, and as arranged the pump will always deliver such quantity of water to the pressure tank D.

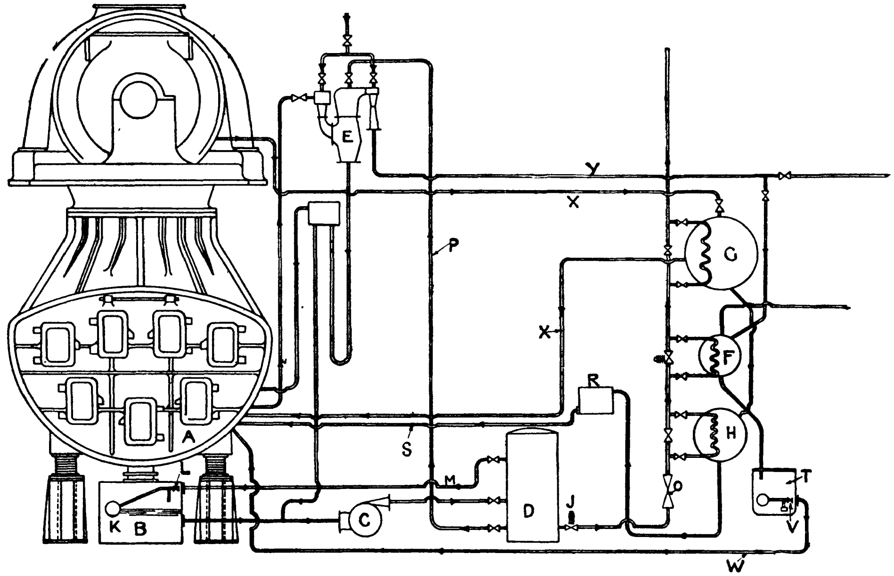

Fig. 131.—Closed Feed System of Messrs. The Mirlees Watson Company

When the steam load falls the water level in the collecting tank B will be maintained by the operation of the float K and valve L, which will allow water to flow back from the pressure tank D to the difference between the lower load and the full load, and the water passing from the pressure tank to the feed heaters will be equal to the amount discharged by the pump, less the amount passing back to the collecting tank B, i.e. equivalent to the condensate corresponding to the actual steam load. This quantity will be registered by the venturi meter O, and will be the steam consumption of the main turbine plus the steam consumption of the ejector air pump E; this latter is a constant known quantity. The steam ejector air pump E is of the two-stage type with intermediate condenser to condense the steam from the first stage. The steam and air from the second stage is discharged to the ejector feed heater F; the condensed water from the ejector condenser is drained through a U-sealing pipe to the condensate pump suction pipe. Alternatively the U-pipe may be dispensed with and a float control box substituted. A steady supply of injection water for the ejector condenser is obtained from the pressure tank D by the pipe P, irrespective of the rate of discharge from the condensate pump. The feed heater G is supplied with steam bled from a low-pressure stage of the main turbine, and the steam condensed in this heater drains to the drain cooler H, where the drain is reduced in temperature before it passes to the main condenser via the vacuum trap R and pipe S. The drain from the ejector heater F gravitates to the drain tank T, and returns to the main condenser through the float-controlled valve V and pipe W. The condensate after passing the venturi meter O flows through the heaters H, F, and G in series before passing to the feed pump suction. By-passes are fitted to these heaters, so that any one may be cut out if desired. An air freeing pipe X connects the heater G with the main condenser, as at low loads the steam supplied to this heater will be under atmospheric pressure, and under this condition the air must be released to the main condenser. An atmospheric connection is fitted to the ejector discharge pipe Y for starting up.

HICK HARGREAVES CLOSED FEED SYSTEM

This system, as designed for land duty, is shown diagrammatically in fig. 132, which has been kindly supplied to the author by Messrs. Hick Hargreaves.

A is the main condenser, and B the water-extraction pump which withdraws the condensate from A and discharges to a closed tank D. The tank D has a small steam supply led to the top of the tank and above the water level, so that a slight steam pressure is maintained above the water level which prevents the ingress of atmospheric air; a leak off is fitted and a slight puff of steam is kept blowing. This arrangement prevents the ingress of air from the outside, but the condensation which takes place on the surface of the water leaves a small blanket of the air which was contained in the steam behind. This air is immediately absorbed by the condensate in the tank, and being a very small quantity, is in some systems neglected.

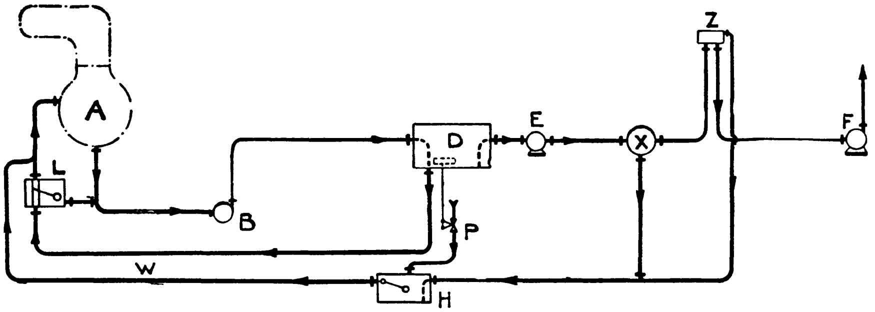

Fig. 132.—Hick Hargreaves Closed Feed System

The water is drawn from the feed tank D by the lift pump E, and discharged through the feed heater or house turbine condenser X, to the elevated closed tank Z. From this tank the water gravitates to the suction of the boiler feed pump F. The tank Z is elevated, so that a positive suction head is given to the feed pump; since the condensate is heated in passing through the feed heater X the elevated position of the tank Z is necessary to enable the feed pumps to deal satisfactorily with the feed water.

Connected to the main condenser is a small tank L which contains a float. The water level in this tank L is the same as that in the extraction pump suction pipe, and rises and falls with it. This float controls a valve to which is connected two pipes, one W to the feed tank, and one to the top of the condenser. When the float falls water is drawn from the feed tank through the pipe W to the float regulated valve, and thence to the top of the condenser; this water does not pass into the tank L, but is by-passed through the float-operated valve. The above arrangement ensures that the water-extraction or condensate pump B is fully charged under all steam loads of the main turbine. Since the float maintains the water level over the condensate pump suction sensibly constant, the pump B will deliver its maximum quantity under all loads, and the excess not drawn by the lift pump E will circulate round pipe W.

From the elevated tank Z an overflow pipe leads to an open auxiliary tank H, which is fitted with a float and valve; from this valve a pipe is led to the top of the main condenser; when the float in the tank H rises the water is drawn up into the condenser. The drains from the heater X are led to the tank H. The make-up feed supply is shown at P, and is controlled by a valve situated in the pipe line, this valve is controlled by a float placed low down in the feed tank, so that when the feed water runs low the float falls and make-up is admitted at P to the auxiliary tank H, and is thereafter passed to the main condenser.

The operation of the system may be described as follows:

When the demand made by the feed pump for water is greater than that supplied from the condenser as condensate, the deficiency is made up from the feed tank D by the water level in it falling.

When the condensate leaving the main condenser is greater than the demand made by the feed pump, the water level rises in the elevated tank Z and overflows through the overflow pipe to the auxiliary tank H. The water level rising in this tank raises the float, thus allowing the excess to be drawn into the main condenser, whereafter it is withdrawn by the condensate pump B and delivered to the feed tank D and accumulates therein.

Fig. 133 shows, in diagrammatic form, the arrangements adopted when an evaporator is installed to provide the necessary make-up supply, and the figure also shows the disposition of venturi meters for the accurate measurement of the condensate.

The diagram is similar to that shown in fig. 132, except that the evaporator is shown at G. The steam supply R to the evaporator is automatically controlled by the float in the feed tank D, so that the output from the evaporator is regulated by the float, and a full feed tank of distilled water is thereby always assured. N and O are venturi meters. The meter O measures the gross quantity of water discharged by the pump B, and the meter N measures the quantity of water by-passed from the tanks D and H. The difference between the readings of the meters O and N gives the quantity

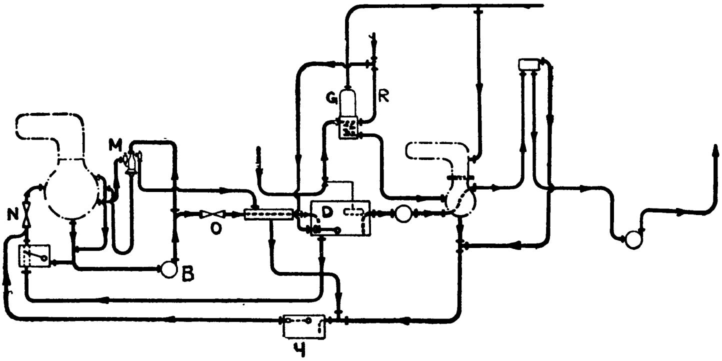

Fig. 133

of steam condensed from the main unit. A small correction is necessary for the quantity of steam used by the first stage of the air ejector M. This steam enters the condensate suction pipe, and is metered along with the main condensate, and should therefore be deducted. The quantity, however, is relatively small and is constant.

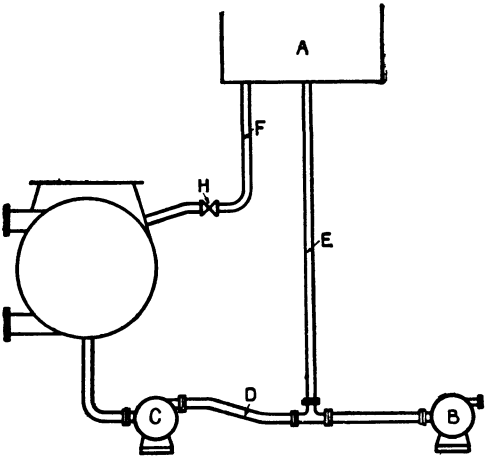

Fig. 134

Another scheme which is sometimes adopted is that shown diagrammatically in fig. 134.

The feed tank A is open and elevated to give a head on the suction of the boiler feed pump B. The water-extraction pump C discharges direct to the feed pump suction, and a vertical pipe E connects the feed suction pipe D with the feed tank. When the pump C is delivering more water than is being demanded by the pump B, the excess flows up the pipe E and into the feed tank A; conversely when B is demanding more water than is being supplied by C the extra water demanded flows down the pipe E from the feed tank and makes up the deficiency; this deficiency thus withdrawn is, as will be seen, not de-aerated, and for this reason the arrangement is sometimes referred to as semi or partially closed. To get over this defect a pipe F, with valve H, is sometimes fitted. The valve H is hand regulated, so that under varying conditions of load, &c., there is always a slight flow up the pipe E.

The arrangement is a simple and flexible one, and for this reason is sometimes adopted.

CHAPTER XII

De-aeration Plants

De-aeration plants are now becoming increasingly popular, and have for their object, as their name implies, the de-aeration of the feed water before it enters the boiler. These plants therefore perform the same function as closed feed systems do. The closed feed system, however, is particularly applicable to new stations where the arrangement of the units can be readily adapted to form a closed system; de-aeration plants, on the other hand, can be readily applied to existing stations, and a great number have been and are being applied to existing stations in this country and abroad, where troubles with corrosion in the feed lines, economizers, and boilers have been experienced. They are also easily worked into the general scheme for new stations. There are three popular types at present on the market.

1. The water to be de-aerated is sprayed into a vessel wherein a vacuum is maintained, and then heated by exhaust steam to boiling-point. With this type the vacuum in the vessel should correspond with the temperature of the water leaving the vessel, that is, the water in the vessel is virtually boiled, so that the air is driven off.

2. The water to be de-aerated is broken up by being allowed to fall over trays arranged inside the de-aerating vessel wherein a vacuum is maintained. Arranged below the trays is a series of horizontal tubes, to the inside of which steam is admitted, the water falling over the hot tubes is thus boiled and broken up, so that the air is effectively liberated. This method is known as the flashing method, that is, the water to be de-aerated is flashed over hot surfaces.

3. The water to be de-aerated is first heated and then injected into a vessel wherein a vacuum is maintained higher than that corresponding to the temperature of the injected water. The water thus injected is superheated; and the energy liberated, due to the reduction in pressure and temperature, causes a violent breaking up, or flashing as it is called, of the feed water, and the air is thus liberated.

This method is greatly in vogue in the United States, where it is manufactured by the Elliott Company.

An example of a de-aerating vessel which employs the first method is shown in fig. 135.

The water enters at A and passes through the spring-loaded spray valve B. A small pressure reduction obtains across this valve so that the water is finely divided and a percentage of the air liberated; this water falls into the perforated cone piece C and flows out through the perforations in the form of a rain. The heating steam enters at D, and passing round the steam belt E, formed by the perforated waist piece F, issues through the perforations and intermingles with the rain of falling water, thus heating the latter by direct contact and driving off the air. The de-aerated water, along with the condensed steam, falls to the bottom and is drawn off at H. The liberated air passes upwards and, meeting the cold spray of entering water, is densified and devaporized and passes out through the branch J, which may be connected to the main condensers of the station or to the suction of an air ejector. Due to the large amount of air cooling and devaporizing which normally obtains with this type of de-aerator, it is usually an easy matter to maintain a vacuum corresponding with the outlet temperature, and under these conditions the de-aeration is quite efficient. This design of heater and de-aerator is that usually adopted on board ship.

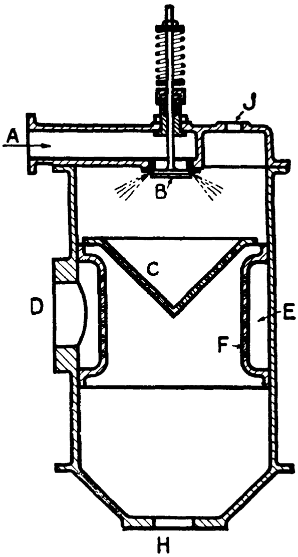

Fig. 135

WEIR DE-AERATING PLANT

Fig. 136 shows an arrangement of de-aerating plant which is an extension of the third method. This arrangement is that now adopted by Messrs. G. & J. Weir, Ltd., Glasgow, for land power stations, and has been found to give good de-aeration.

The condensate being discharged by the water-extraction pumps is not discharged direct to the feed tank A but goes to the de-aerating plant at B and passes first through a de-vaporizer C, which is a tubular apparatus very similar to a steam condenser; the condensate passes down through the inside of the tubes, and, leaving at the bottom, passes to the air ejector D by the pipe E; in flowing through the air ejector it condenses the steam used by the jets and is itself raised in temperature; thereafter it passes by the pipe F to the feed tank A. Auxiliary exhaust or live steam is led to the feed tank by the pipe G and nozzle heater H to give a feed tank temperature of 140° to 160° F.

The feed water leaves the feed tank by the pipe J and enters the first stage chamber K through the butterfly throttle valve L, which is controlled by the hollow copper float M.

Beneath the first stage chamber is a second stage chamber N, having a similar inlet butterfly throttle valve O and float P; the feed water flows from the chamber K to the chamber N through the valve O, and collects at the bottom of the chamber N, whence it is withdrawn by the water-extraction or booster pump Q and delivered direct to the suction of the boiler feed pump R, and thence to the boilers.

The principle of this de-aerator is as follows:

The heating is done in the feed tank under atmospheric pressure, this drives off a portion of the air; a vacuum is maintained in the vessel N by the air ejector D, the vacuum temperature being 20° F. higher than the temperature of the water entering the first stage chamber. A vacuum a few inches Hg lower is maintained in the first stage chamber. The water therefore passing the first stage valve L receives a pressure drop which liberates a portion of the air and also some vapour, this air and vapour pass to the second stage chamber by way of the pipe S and enter above a perforated blast plate T; some air and vapour is also liberated at the second throttle valve O, and this, together with the air and vapour liberated at the first stage, blasts through the perforated plate T along with the feed water and causes a violent disintegration of the water which practically frees the water of all

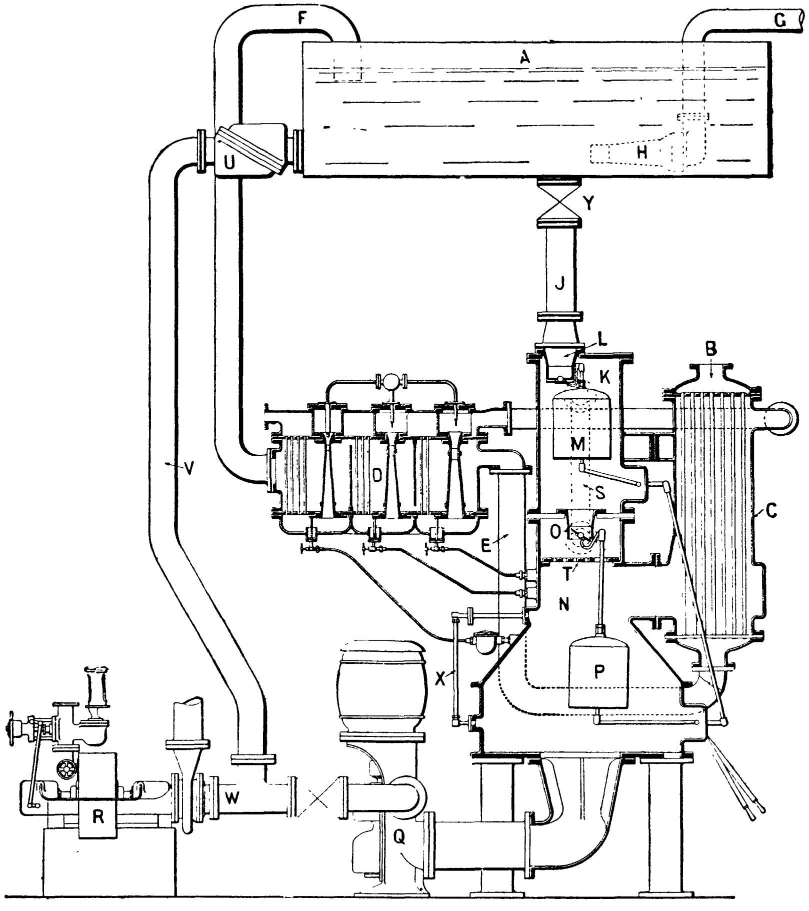

Fig. 136.—Weir De-aerating Plant

air. Figures as low as zero oxygen have been obtained from these plants.

A normal set of temperature conditions for these plants would be as follows:

The condensate enters C at 70° F. and leaves C at 90° F., passes through the air ejector and leaves at 94° F. Sufficient steam enters the feed tank to give a tank temperature of 160° F.; in passing through the first and second stage chambers the liberated vapour lowers the temperature an amount exactly equal to the gain in temperature of the condensate passing through the de-vaporizer, that is, 20° F., so that the water leaving the de-aerator will be 140° F.

A non-return valve U and pipe V are arranged as shown, the valve U closing upwards and the pressure in the pipe W created by the booster pump Q is arranged higher than that corresponding to the height of the feed tank. Under working conditions, therefore, the pressure in W holds the non-return valve W shut and all the water passes to the feed pump; should, however, the booster pump stop from any cause whatever, the pressure in the pipe W falls, thus allowing valve U to open and the boiler feed pump automatically draws direct from the feed tank.

To start up this plant the condensate is made to circulate the de-vaporizer C and air ejector D first; steam is then turned on to the air ejector and vacuum created in the de-aerating chambers, valve Y is then opened and water flows into the first and thereafter into the second stage chambers and comes under the control of the float-controlled valves L and O. The booster pump thereafter is started, and lastly the boiler feed pump, if it be not already running and drawing direct from the feed tank; if the latter condition holds, the pressure created by the booster pump will close valve U and the change over from the feed tank to the de-aerating plant will take place automatically.

The performance of these plants when measured in cubic centimetres of oxygen per litre of water remaining in the water after treatment varies with the temperature of the water leaving the de-aerator, the variation being approximately from 0·15 at 100° F. to 0·03 at 140° F.

HICK HARGREAVES DE-AERATING PLANT

Another design of de-aerating plant which employs the first method is that manufactured by Messrs. Hick Hargreaves & Co., Bolton.

The arrangement is shown in fig. 137 and consists essentially of a surface preheater C, a de-aerating vessel A and an air ejector B. The water enters from the hotwell at E and makes two passes through the preheater C, wherein it is heated to a predetermined temperature, thereafter it is sprayed upwards into the de-aerator A through the nozzles F in the form of a divided spray, and collects in the bottom of the vessel A. A float-controlled throttle valve G is fitted at the inlet to spray nozzles to maintain a water-level over the extraction-pump suction. The air ejector B withdraws the liberated air and delivers it along with its own operating steam to the preheater C. A collector with a float-controlled valve H is arranged under the preheater C to collect the drain water from the heater, the valve H allowing same to flow to the de-aerator by the pipe J to de-aerator without air being drawn into the latter and impairing the vacuum.

The manufacturers of this design of plant claim that they have found from experiment that there is a critical temperature at which the maximum degree of de-aeration can be obtained, when the water is sprayed into a vacuum corresponding to this temperature. This phenomenon was mentioned on p. 240 when we were dealing with surface feed water heaters, and the critical temperature mentioned by them is presumably that mentioned on p. 240, namely 176° F. To admit of the critical temperature being maintained a thermostatic control valve is fitted to control the quantity of steam admitted to the heater; the steam supply to the heater is that from the air ejector augmented by the available exhaust steam on the station or by direct boiler steam.

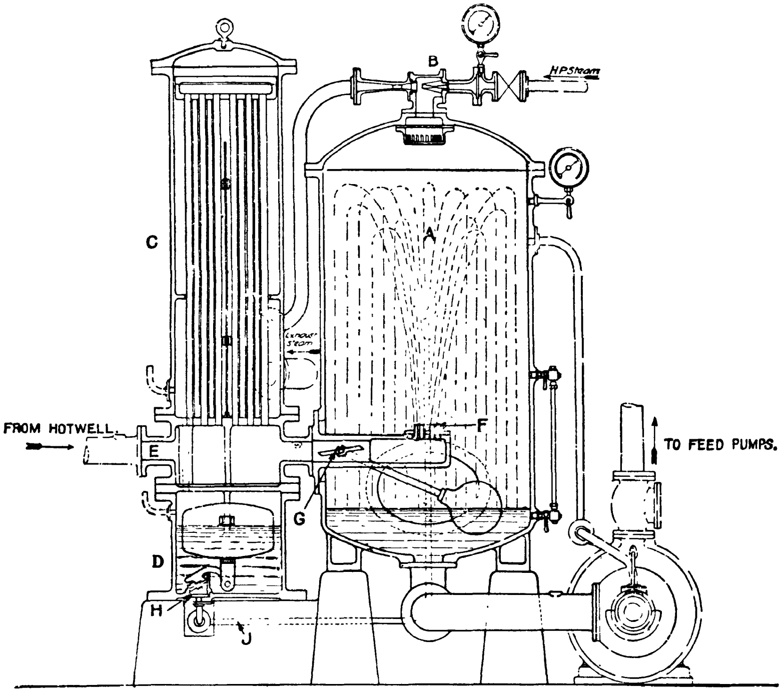

Fig. 137.—Hick Hargreaves De-aerating Plant

These de-aerators are guaranteed by the manufacturers to reduce the oxygen content to less than 0·1 c. c. of oxygen per litre of water, and to reduce the carbon dioxide to practically nothing.

CONTRAFLO ENGINEERING COMPANY'S DE-AERATING PLANT

An example of a de-aerator which employs the second method is shown in fig. 138, and is the design adopted by the Contraflo Engineering Company, Ltd., 3 Central Buildings, Westminster, London.

The water to be de-aerated flows from the feed tank A through the float-operated valve B to the top of the de-aerator. The water falls over a series of trays C, which give a preliminary breaking up to the water and free a percentage of the air; the water leaves the bottom tray in the form of a rain and falls on hot tube surfaces D. These surfaces are maintained hot by exhaust steam which enters at E and flows through the interior of the tubes. The exhaust steam condensed leaves at F, and may be passed to the main condenser. G is an air ejector which maintains the vacuum in the de-aerator. This vacuum should correspond to the temperature of the water leaving the de-aerator. Due to the water falling on the hot surfaces, a portion of this water will be vaporized. The vapour rises upwards through the water falling over the trays and imparts its latent heat to the water; the combined effect of this heating, together with the breaking up of the water, liberates the first portion of the air; the remaining portion is liberated on the hot surfaces by the violent ebullition of the water passing down from one hot tube to the other. These de-aerators are claimed to give a high degree of de-aeration, the makers claiming to completely remove all traces of dissolved oxygen. The water is withdrawn from the de-aerator by the pump H, the suction head on which is maintained constant by the float J and valve B.

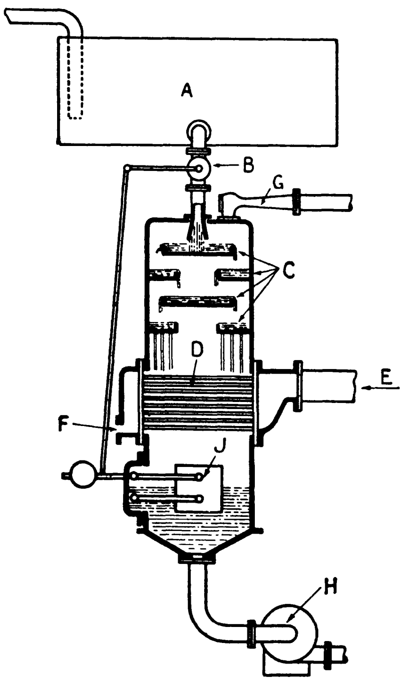

Fig. 138

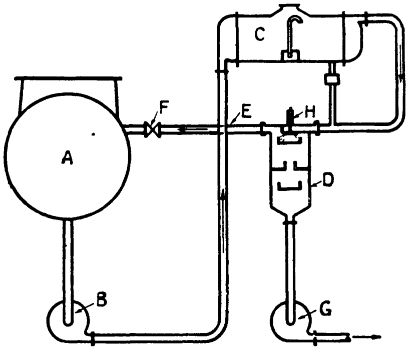

Fig. 139

An example of the third method is shown in fig. 139, the arrangement shown being one due to the Contraflo Engineering Company.

A is the main condenser from which the water-extraction pump B draws and delivers the condensate through the tubes of the surface heater C to the inlet of the de-aerator D. Exhaust steam is supplied to the heater C, so that the condensate is heated on its way to the de-aerator D.

A vacuum greater than that corresponding to the inlet water temperature is maintained in the de-aerator by the latter being connected to the condenser by the pipe E and regulating valve F. The de-aerator is fitted with a spring-loaded spray valve H, across which the inlet water flashes to liberate the air; the air, together with the flashed vapour or steam, passes along the pipes E to the condenser A, where the vapour is condensed and the air is extracted by the main air pump fitted to the condenser A. The de-aerated water is withdrawn from the de-aerator by the pump G. To ensure efficient separation of the dissolved air, the temperature drop or gradient across the spray valve H should not be less than 22° F., that is, if the water entering the de-aerator be at a temperature of 160° F. the vacuum carried in the de-aerator should be that corresponding to $160 - 22 = 138°$ F., or $24\frac{1}{2}$ in.

METHOD OF MEASURING DISSOLVED OXYGEN IN FEED WATER

Many power station engineers are now demanding a guarantee from makers of de-aerating plants for the oxygen content of the water as it leaves the de-aerator. The usual guarantee varies from 0·2 to 0·05 c.c. of oxygen per litre of water and the test is usually made by the iodometric method, due to Winkler. The test may be performed as follows:

To 250 c.c. of the water add 2 c.c. of manganous sulphate solution and 2 c.c. of a solution containing potassium iodide and sodium hydrate. Mix gently. Add 3 c.c. of dilute sulphuric acid and mix. Titrate with N/40 sodium thiosulphate solution, using starch solution as indicator.

1 c.c. N/40 thiosulphate solution = 0·56 c.c. oxygen per litre.

The strengths of the various solutions and the method of standardizing the thiosulphate solution are given in any modern textbook on water analysis.

METHOD OF DETECTING THE CORROSIVE PROPERTIES OF FEED WATER

Messrs G. & J. Weir, Ltd., have recently placed on the market an apparatus for testing the corrosiveness of feed water, and called by them a corrosion detector. This instrument does not give a quantitive measure of the dissolved gases in the feed water, it rather detects whether the feed water is corrosive or not, and to a certain extent the degree of corrosiveness of the feed water. The essentials of the instrument are a polished steel plate or disc, over the polished surface of which the feed water to be tested is made to flow; the back of the disc is heated by any convenient means. The heat flow across the disc to the polished surface liberates the dissolved gases in the feed water on the polished surface, which is immediately attacked by the liberated gases.

Fig. 140 shows a section through the instrument.

The water inlet is connected to the feed pipe and the water outlet is led to waste. The heat is applied to the back of the steel disc by connecting the steam inlet to any convenient steam or exhaust pipe, and the drain is led to waste. In front of the polished plate a glass plate is fitted through which the action may be observed going on. To make a test the instrument is connected up as above described; the water inlet valve is first opened and the water allowed to flow freely over the polished surface for a few moments to expel all air. When all air is expelled the water inlet valve is throttled down till a small flow takes place, which may be observed at the open end of the water outlet pipe. The steam inlet valve is then opened full, and if the water is corrosive the liberated gases may be seen adhering to the polished surface, and in a few moments the plate will be observed to be quite discoloured; if the water is non-corrosive the bright polish on the plate will be unaffected. The makers recommend a standard time of one hour for each test, this enables a comparison of the corrosiveness of feed water to be obtained by observing the depth and extent of the pit marks

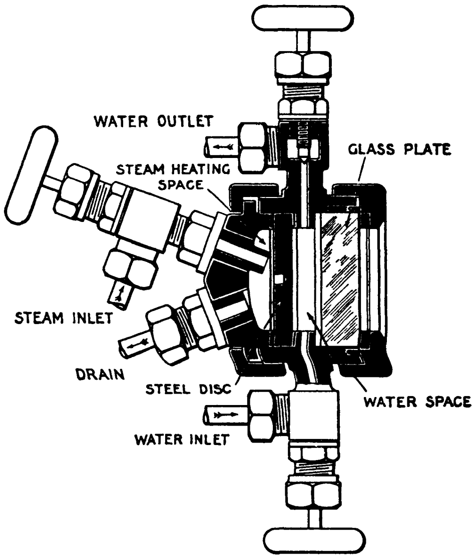

Fig. 140

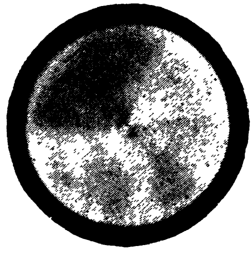

Fig. 14

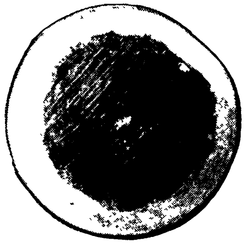

Fig. 142

on individual discs. It is worthy of note that this instrument detects corrosion whether the action is due to oxygen or carbonic acid gas.

Fig. 141 and 142 are reproductions of two discs after two hours' treatment. Fig. 141 shows the condition of the disc after exposure to feed water in a closed feed system, while fig. 142 shows the condition after exposure of feed water in an open feed system. The water used for both these discs was pure distilled water, and the necessity for de-aeration is clearly shown.

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